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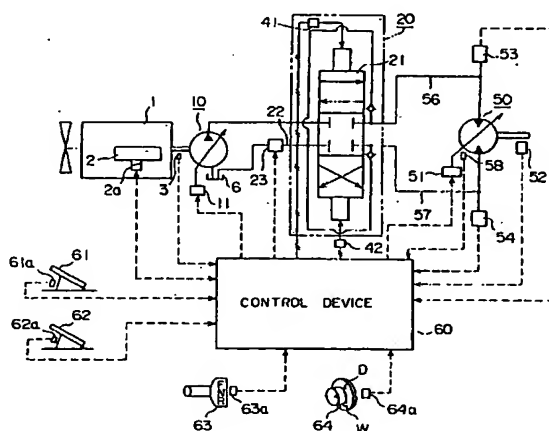
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(54) Speed change control method of hydraulic driving apparatus for vehicles and speed changing device

(57) The present invention relates to a speed change control method of a hydraulic driving apparatus for vehicles and a speed changing device which provide excellent running efficiency and controllability. In order to achieve the above-described advantages, the speed change control method comprises discriminating between at least a power running and a brake running from

said accelerating amount ( $\theta$ ) and the rotation speed ( $\omega_{out}$ ) of said hydraulic motor (50); and allowing return oil from said hydraulic motor (50) to a tank (6) to have a high pressure so as to be fed to the supply side of said hydraulic motor (50) if inlet pressure ( $P_p$ ) to said hydraulic motor (50) is lower than permissible suction pressure of said hydraulic motor (50) when discriminating the brake running.

FIG. 1



## Description

### TECHNICAL FIELD

[0001] The present invention relates to a speed change control method of a hydraulic driving apparatus for vehicles and a speed changing device, and more particularly, to a speed change control method of a hydraulic driving apparatus for use in construction vehicles, agricultural machinery and automobiles, etc., and a speed changing device.

### BACKGROUND ART

[0002] Hitherto, various types of power transmission devices for use in construction vehicles, agricultural machinery and automobiles, etc. such as of mechanical type, hydraulic type and electric type have been proposed and used. Regarding small vehicles for construction equipment, those of hydraulic type have been used relatively often. This is because those of hydraulic type can change their running speed from zero to infinity, and a merit of excellent operability has been highly regarded. On the other hand, those of hydraulic type have disadvantages of lower efficiency and higher cost to those of mechanical type. However, operating machines for digging and earth moving, etc. are mounted in construction equipment such as a wheel hydraulic excavator, and all power from an engine is converted by a hydraulic pump in order to actuate the operating machines, so that the use of hydraulic driving apparatuses may become less expensive conversely. When such hydraulic driving apparatuses are used, there included two types: closed circuit and open circuit. They have different characteristics, and they are used in accordance to purpose.

[0003] For example, in construction equipment used mainly for the purpose of running, when large amounts of flowing oil pressure is required, those of open circuit type are used. Closed center load sensing circuits have been used in operating machines in terms of improvement in operability, and closed center valves are adopted therein. On the other hand, when running efficiency or controllability is emphasized, those of closed circuit type are used, and a running hydraulic pump and an operating machine hydraulic pump are used therein.

[0004] In addition, a circuit using a counter balance valve shown in Fig. 37 has been known and it controls return oil from the counter balance valve to effect speed control (runaway prevention) when descending a slope. This construction comprises a variable displacement hydraulic pump 210 driven by a driving source 1 such as an engine; a capacity control device 211 for controlling capacity of the hydraulic pump 210; a forward-reverse directional control valve 212; solenoid operated proportional valve 213 (forward) and 214 (reverse) for controlling the forward-reverse directional control valve 212; a counter balance valve 215 connected to the forward-reverse directional control valve 212; a variable

displacement hydraulic motor 216 for receiving the pressure oil from the counter balance valve 215; and a capacity control device 217 for controlling the capacity of the hydraulic motor 216.

[0005] However, the above-described conventional circuit encounters the following problems.

i) The use of the counter balance valve in a running circuit reduces efficiency because the valves are controlled by restriction. Moreover, since heat is generated while running, a large cooler and an engine of larger outputs are required, so that size and cost of a vehicle increases.

ii) The use of the closed center valve also in the running circuit of a vehicle emphasizing workability encounters a similar malfunction because the valve is controlled by restriction similar to the counter balance valve. Particularly, in a high-speed, long-distance running vehicle, resistance increases reducing efficiency and heating value also increases, so that a large cooler is required.

iii) The gradients of changes of the engine rotation speed are taken on the same line during low speed and during high speed, so that follow-up properties of the engine with respect to a depressing amount of the brake pedal is bad or oversensitive during either of low speed and high speed.

iv) During forward F, or reverse R, if a shock at the time of starting is decreased, or slow speed forwarding is made possible, starting properties deteriorate. If the starting properties are improved, the shock increases, or the slow speed forward cannot be effected.

v) The gradients of changes of an inclination-rotation angle of the hydraulic motor are taken on the same line during low speed and during high speed, so that the ratio of the change decreases to deteriorate follow-up properties at the time of high speed, and hunting occurs at the time of low speed.

vi) The gradients of changes of the inclination-rotation angle of the hydraulic motor are held constant with respect to a brake depressing angle, so that follow-up properties deteriorate when the brake depressing angle is large. On the other hand, when the brake depressing angle is small, the brake becomes too effective, providing greater shock.

vii) If an operating lever is operated at the time of running, the operating machine is actuated, thereby incurring dangers.

viii) The discharge capacities of the hydraulic pump are equalized during forwarding and during reversing, so that driving torque at the time of reversing for escaping from uneven ground, etc. is insufficient.

ix) A plurality of hydraulic pumps are used, each of driving apparatuses are required therefor, so that the space increases.

## DISCLOSURE OF THE INVENTION

[0006] The present invention has been made to solve the drawbacks of the prior art, and its object is to provide a speed change control method of a hydraulic driving apparatus for vehicles and a speed changing device which have excellent running efficiency and controllability.

[0007] In a first aspect of a speed change control method of a hydraulic driving apparatus for vehicles according to the present invention, there is provided a speed change control method of a hydraulic driving apparatus for vehicles which operates a shifter to select forward and reverse of a vehicle, changes rotation speed of an engine by an accelerating amount, and supplies pressure oil from a hydraulic pump driven by the engine to a hydraulic motor through a directional control valve so as to control rotation speed of the hydraulic motor for running the vehicle wherein said method comprising: discriminating between at least a power running and a brake running from said accelerating amount and the rotation speed of said hydraulic motor; and allowing return oil from said hydraulic motor to a tank to have a high pressure so as to be fed to the supply side of said hydraulic motor if inlet pressure to said hydraulic motor is lower than permissible suction pressure of said hydraulic motor when discriminating the brake running.

[0008] In an aspect of the speed changing device, there is provided a speed changing device of a hydraulic driving apparatus for vehicles including an engine of which rotation speed changes in response to an accelerating amount, a hydraulic pump driven by the engine, a brake pedal, and a hydraulic motor for receiving pressure oil from the hydraulic pump through a directional control valve to run a vehicle, said device comprising: motor pressure sensors for detecting inlet pressure to said hydraulic motor and outlet pressure from said hydraulic motor; a braking amount detection sensor for detecting a braking amount of said the brake pedal; a variable pressure two-stage back pressure valve for controlling return pressure of a return circuit formed between said directional control valve and an oil tank; and a control device, wherein said control device, at the time of the braking, compares said detected inlet pressure with permissible suction pressure of said hydraulic motor, and outputs a command to said two-stage back pressure valve to increase said return pressure when said detected inlet pressure is lower than said permissible suction pressure.

[0009] According to such a construction of the present invention, a counter balance valve is not used in a running circuit, and the directional control valve is controlled by restriction only when it is required, so that heating while running is reduced. This eliminates the need for a large cooler and also the need for the engine of larger outputs, so that the vehicle can be reduced in size and fuel economy is improved. The cost of the vehicle is also reduced. In addition, the engine rotation speed and the

discharge capacity of the hydraulic motor are changed separately during low speed and during high speed, and they are changed with a predetermined range between low speed and high speed, separate controls can be effected during low speed and during high speed, so that follow up properties of the engine are improved. As the hydraulic motor is constructed similarly, follow-up properties with respect of the required torque amount are excellent, and occurrence of hunting can be prevented even if the vehicle speed is low. Further, at the time of braking, if the brake depressing angle is large, the discharge capacity of the hydraulic motor is increased to improve follow-up properties, so that the brake becomes easily effective. On the other hand, if the brake depressing angle is small, braking can be slowed down to effect slight movement braking. In addition, the discharge capacity of the hydraulic motor while reversing is increased larger than that of while forwarding to provide large output torque, escape from uneven ground, etc. is facilitated.

[0010] Further, by selecting the operation mode, improvement in operability, which is a merit of the closed center load sensing circuit, can be obtained. On the other hand, when the running mode is selected, resistance and heating value can be reduced by opening the closed center valve, and improvement in running efficiency can be achieved. Particularly, in the vehicle having the operating machine, by using a hydraulic pump of the operating machine also in the running circuit, only one hydraulic pump is sufficient, so that space and cost of the vehicle can be reduced. In addition, as a measure to the nonuse of the counter balance valve, discharge side return oil from the hydraulic motor is sent to a supply side pipe from the hydraulic pump to the hydraulic motor through suction valves, thereby preventing occurrence of cavitation.

[0011] Further, a running mode and an operation mode are provided, and at the time of the running mode, operation can be safely performed without operation of the operating machine even if the operating lever is operated, and the operating machine takes priority at the time of the operation mode, so that the vehicle with good workability can be provided.

## BRIEF DESCRIPTION OF DRAWINGS

## [0012]

Fig. 1 illustrates a concept of a speed changing device of a hydraulic driving apparatus for vehicles according to a first embodiment of the present invention;

Fig. 2 is a detail circuit diagram of Fig. 1;

Fig. 3 is a flowchart of an operation mode switching of the hydraulic driving apparatus according to the first embodiment;

Figs. 4 to 6 are flowcharts of a series of controls when selecting a running mode according to the first

embodiment;

Figs. 7 to 18 are diagrams in the running mode according to the first embodiment;

Fig. 7 is a diagram showing the relationship among the engine rotation speed, the engine torque and the discharge amount of a hydraulic pump;

Fig. 8 is a diagram showing the relationship among the accelerating amount, the engine rotation speed, and the vehicle speed;

Fig. 9 is a diagram showing the relationship among the vehicle speed, the opening degree command value of a running valve, and the accelerating amount;

Fig. 10 is a diagram showing the relationship between frequency of a low-pass filter through which a flow rate command value is passed and the gain;

Fig. 11 is a diagram showing the relationship between the inlet pressure to a hydraulic motor and the inclined shaft command value of the hydraulic motor on running;

Fig. 12 is a diagram showing the relationship between the rotation speed of the hydraulic motor and the inclined shaft command value of the hydraulic motor on braking;

Fig. 13 is a diagram showing the relationship between frequency of a low-pass filter through which the inclined shaft command value is passed and the gain;

Fig. 14 is a diagram showing the relationship between the forward flow rate command value and the current of a forward solenoid operated directional control valve;

Fig. 15 is a diagram showing the relationship between the reverse flow rate command value and the current of a reverse solenoid operated directional control valve;

Fig. 16 is a diagram showing the relationship between the inclined shaft command value of the hydraulic motor and the current of a hydraulic motor solenoid operated directional control valve;

Fig. 17 is a diagram showing the relationship between the absorbing torque command value to a TVC valve and the current;

Fig. 18 is a diagram showing the relationship between the engine rotation speed command value and the current to a solenoid;

Figs. 19 to 21 are flowcharts of a series of controls when selecting an operation mode according to the first embodiment;

Figs. 22 to 34 are diagrams in the operation mode according to the first embodiment;

Fig. 22 is a diagram showing the relationship among the engine rotation speed, the engine torque and the discharge amount of a hydraulic pump;

Fig. 23 is a diagram showing the engine rotation speed with respect to the engine rotation setting;

Fig. 24 is a diagram showing the relationship among the accelerating amount, the accelerating correc-

tion amount, and the engine rotation speed;

Fig. 25 is a diagram showing the relationship among the vehicle speed, the opening degree command value of a running valve, and the accelerating correction amount;

Fig. 26 is a diagram showing the relationship between frequency of a low-pass filter through which the flow rate command value is passed and the gain;

Fig. 27 is a diagram showing the relationship between the inlet pressure to a hydraulic motor and the inclined shaft command value of a motor;

Fig. 28 is a diagram showing the relationship between the accelerating amount and the inclined shaft command value of a motor on braking;

Fig. 29 is a diagram showing the relationship between frequency of a low-pass filter through which the inclined shaft command value on braking is passed and the gain;

Fig. 30 is a diagram showing the relationship between the forward flow rate command value and the current of the forward solenoid operated directional control valve;

Fig. 31 is a diagram showing the relationship between the reverse flow rate command value and the current of the reverse solenoid operated directional control valve;

Fig. 32 is a diagram showing the relationship between the inclined shaft command value of the hydraulic motor on braking and the current of the hydraulic motor solenoid operated directional control valve;

Fig. 33 is a diagram showing the relationship between the absorbing torque command value to the TVC valve and the current;

Fig. 34 is a diagram showing the relationship between the fuel injection amount command value and the solenoid current;

Fig. 35 is a hydraulic circuit diagram of a speed changing device of a hydraulic driving apparatus according to a second embodiment of the present invention;

Fig. 36 is a hydraulic circuit diagram of a speed changing device of a hydraulic driving apparatus according to a third embodiment of the present invention; and

Fig. 37 is a schematic hydraulic circuit diagram showing a speed changing device of a hydraulic driving apparatus using a counter balance valve according to the prior art.

#### BEST MODE FOR CARRYING OUT THE INVENTION

[0013] The preferred embodiments of a speed change control method of a hydraulic driving apparatus for vehicles and a speed changing device according to the present invention will be described in detail with reference to the attached drawings.

[0014] Referring to Fig. 1, a variable displacement hydraulic pump 10 (hereinafter, referred to as a hydraulic pump 10) driven by an engine 1 sucks oil therein from an oil tank 6, converts outputs of the engine 1 into pressure oil, and feeds the pressure oil to a variable displacement hydraulic motor 50 (hereinafter, referred to as a hydraulic motor 50) via a switching device 20. The hydraulic motor 50 converts the pressure oil from the hydraulic pump 10 into rotary torque so as to drive a construction vehicle, etc. Such a hydraulic driving apparatus has already been known. To the engine 1, a fuel injection pump 2 for supplying fuels to the engine 1 is attached, and a solenoid 2a, for example, is attached to the fuel injection pump 2. The solenoid 2a receives a command from a control device 60 such as a controller, etc. to control the fuels supplied to the engine 1, and control rotation speed and outputs of the engine 1.

[0015] To the hydraulic pump 10, a servo device 11 which can allow absorbing torque to be variable is attached, and to the hydraulic motor 50, a servo device 51 which controls a swash plate, etc. for allowing displacement to be variable is attached. To the servo devices 11, 51, pressure oil is supplied via a solenoid operated directional control valve, etc. which are switched upon receipt of a command from the control device 60. The servo devices 11, 51 switch and control the pressure oil, and control absorbing torque of the hydraulic pump 10 and a discharge capacity of the hydraulic motor 50.

[0016] A running closed center load sensing valve 21 (hereinafter, referred to as a running valve 21) of the switching device 20 comprises three positions, and pilot pressures from solenoid operated directional control valves (solenoid operated proportional control valves) 41, 42, which are provided at both ends, act thereon to switch the valve 21 to a forward position or a reverse position. The solenoid operated directional control valves 41, 42 are switched upon receipt of a command from the control device 60. A return circuit 22 formed between the running valve 21 and the oil tank 6 is provided with a two-stage back pressure valve 23, which is switched in two stages by the command from the control device 60, and controls return pressure of return oil to the tank 6.

[0017] The control device 60 is provided with an engine rotation speed sensor 3 for detecting rotation speed of the engine 1, a motor rotation speed sensor 52 for detecting a rotation speed of the hydraulic motor 50 to detect a vehicle speed, and motor pressure sensors 53, 54 for detecting inlet and outlet pressures at the side of a pipe 56 or a pipe 57 which flow in and flow out of the hydraulic motor 50. To the hydraulic motor 50, an angle sensor 58 for measuring inclination-rotation angle  $\alpha$  of a swash plate, etc. is attached to detect braking capacity of a retarder for braking a vehicle. Here, in order to make it understandable, for reasons of convenience, the motor pressure sensor 53 secured to the pipe 56 on which pressure acts in the direction of forward rotation of the

hydraulic motor 50 is referred to as a forward pressure sensor 53, and the motor pressure sensor 54 secured to the opposite side pipe 57 is referred to as a reverse pressure sensor 54. Incidentally, although the motor rotation speed sensor 52 detects the rotation speed and direction, two sensors may be provided to detect the rotation speed and the direction, separately.

[0018] In addition, the control device 60 is provided with an accelerator pedal 61 for controlling a vehicle speed, a brake pedal 62 for controlling a vehicle braking, a shifter 63 for selecting a vehicle travelling direction (referred to as forward F, reverse R, and neutral N), and a mode selection switch 64. The mode selection switch 64 is the switch which actuates an operating machine switching device 45 for driving an operating machine (not shown), and selects an operation mode W for the purpose of mainly operating (including an operation while running), or a running mode D only for effecting vehicle running. The control device 60 is provided with an engine rotation setting dial 66 for setting rotation speed of the engine 1 during operation.

[0019] An accelerating amount detection sensor 61a for detecting an accelerating amount is attached to the accelerator pedal 61, a braking amount detection sensor 62a for detecting a braking amount is attached to the brake pedal 62, a shifter selected position sensor 63a for detecting a vehicle travelling direction is attached to the shifter 63, and a mode detection sensor 64a for detecting the operation mode W and the running mode D is attached to the mode selection switch 64, respectively.

[0020] Each of the above-described sensors are connected to the control device 60, and each of predetermined signals are inputted to the control device 60 therefrom.

[0021] Referring to Fig. 2, a pipe 12 is secured to a hydraulic pump 10, the pipe 12 is branched, and one pipe 12a is connected to a vehicle driving force switching device 20. The other pipe 12b is further branched into pipes 12c, 12d, 12e, ... To the pipe 12c, an unload valve 13 is connected, and to the pipes 12d, 12e, ..., a plurality of operating machine closed center load sensing valves 46a, 46b... (hereinafter, referred to as operating machine CLSS valves 46a, 46b...) are connected. The operating machine CLSS valve 46a supplies and discharges pressure oil to and from operating machine actuators 47, 47a.

[0022] The servo device 11 of the hydraulic pump 10 is composed of a torque value control valve 11a (hereinafter, referred to as a TVC valve 11a), a load sensing valve 11b (hereinafter, referred to as an LS valve 11b), and a servo 11c. To the TVC valve 11a, a pilot tube 12w from the pipe 12, and the control device 60 are connected. The TVC valve 11a has a power mode function such that it receives a command from the control device 60, widely controls discharge pressure of the hydraulic pump 10 to send to the LS valve 11b, and widely and freely controls a discharge amount of the hydraulic

pump 10.

[0023] For example, when a command current from the control device 60 is small, the pump discharge amount is increased to the maximum, and when the command current is large, the pump discharge amount is reduced, thereby effecting a constant horsepower control, etc. for holding pump discharge capacity  $\times$  discharge pressure constant. To the LS valve 11b, a pilot tube 12w from the pipe 12 and a tube 48w of an LS circuit are connected. The LS valve 11b receives the discharge pressure PP of the hydraulic pump 10, and the highest pressure PS among pressures from the circuit (LS circuit) of the running valve 21 and operating machine CLSS valves 46a, 46b... through check valves 49, 49a. The LS valve 11b controls pressure from the TVC valve 11a to send to the servo 11c so that an LS differential pressure ( $PLS = PP - PS$ ), which is the differential pressure ( $PP - PS$ ) therebetween, is held constant, thereby controlling an angle of the swash plate (the discharge amount of the pump) of the hydraulic pump 10.

[0024] The vehicle driving force switching device 20 comprises the running valve 21, a pressure compensation valve 21a, suction valves 31, 32, and safety valves 33, 34. To the running valve 21, suction valves 31, 32 and safety valves 33, 34 are attached. Each one side of the suction valve 31, 32 and the safety valve 33, 34 is connected between the running valve 21 and the hydraulic motor 50, and each other side thereof is connected between the running valve 21 and the two-stage back pressure valve 23 by means of the return circuit 22 from the running valve 21. Incidentally, the suction valves 31, 32, and the safety valves 33, 34 may be mounted to pipes without being attached to the running valve 21.

[0025] In addition, at both ends of the running valve 21, pilot pressure for switching to forward or reverse acts on predetermined both end surfaces thereof so as to switch the running valve 21 in response to an operation of the shifter 63 (Fig. 1). The pilot pressure is generated in such a manner that discharge pressure of a non-illustrated pilot pump receives a command from the control device 60, and the forward solenoid operated directional control valve 41 or the reverse solenoid operated directional control valve 42 is switched, whereby a predetermined pressure responsive to a command is generated.

[0026] A motor solenoid operated directional control valve 55 is switched upon receipt of a command from the control device 60, and pressure of the pilot pump acts on the servo device 51 so as to control displacement of the hydraulic motor 50. The hydraulic motor 50 receives the discharge amount from the hydraulic pump 10 in response to an accelerating amount so as to output a predetermined rotation speed, i.e., a predetermined vehicle speed.

[0027] The solenoid operated directional control valve 24 is switched upon receipt of a command from the control device 60, and pressure of the pilot pump acts on the two-stage back pressure valve 23 so as to control return pressure of return oil to the tank 6 in the two steps

of high pressure and low pressure. When high pressure, the discharge side return oil from the hydraulic motor 50 is fed to either of the supply side pipes 56 and 57 through the suction valve 31 or 32. When low pressure, the return oil is mainly returned to the oil tank 6. At this time, a part of return oil is fed as needed to either of the pipes 56 and 57 of the supply side from the hydraulic pump 10 to the hydraulic motor 50.

[0028] Regarding an operation due to the above-described construction, a mode selection for operating a hydraulic driving apparatus will be first described.

[0029] In a non-illustrated construction vehicle having an operating machine, an operation of a hydraulic driving apparatus comprising the hydraulic pump 10, the running valve 21 and the hydraulic motor 50 is selected by a mode selection switch 64. Referring to a flow of Fig. 3, when the mode selection switch 64 is selected to a hydraulic driving control of the running mode D in step 301, a procedure advances to step 302. In step 302, the running mode D is detected by the mode detection sensor 64a, and a signal of the running mode D is sent to the control device 60. In the control device 60, a map of the running mode D and judgement are accessed. That is, a command signal of the running mode D for only effecting running is outputted from the control device 60. For example, when a non-illustrated operating machine lever is operated to operate the operating machine, the operating machine CLSS valve 46a is controlled so as not to be actuated.

[0030] In step 301, when the mode selection switch 64a is selected to a hydraulic driving control on operation (operation mode W), the procedure advances to step 303. In step 303, similar to step 302, a map of the operation mode W and judgement are accessed, and a command signal of the operation mode W mainly for the purpose of operating is outputted. For example, in performing operation while running, if the running valve 21 and the operating machine CLSS valves 46a, 46b... are operated simultaneously, a command signal mainly for the purpose of supplying to the operating machine CLSS valves 46a, 46b... is outputted from the control device 60 when less amount is discharged from the hydraulic pump, while a command signal for supplying to the running valve 21 and the operating machine CLSS valves 46a, 46b... is outputted from the control device 60 when large amount is discharged.

[0031] Next, a hydraulic driving control when the running mode D is selected will be described with reference to Fig. 4, Fig. 5 and Fig. 6. Incidentally, a control of discharge capacity of the hydraulic motor 50 on running and on retarding (braking), and switching operation of the two-stage back pressure valve 23 will be described.

[0032] In step 311, various signals are inputted to the control device 60. That is, the control device 60 reads in ever-changing signals from the various sensors. The signals include a change of the shifter position (a change of forward F, reverse R, neutral N) from the shifter selected position sensor 63a, an engine rotation

speed  $\omega_e$  from the engine rotation speed sensor 3, an accelerating amount  $\theta$  from the accelerating amount detection sensor 61a, a motor rotation speed  $\omega_{out}$  (i.e., vehicle speed  $V$  of a vehicle) from the motor rotation speed sensor 52, inlet and outlet pressures  $P_{pc}$  from the motor pressure sensors 53, 54 to the hydraulic motor 50, and a braking capacity  $R_c$  of the retarder from the angle sensor 58. Among the inlet and outlet pressures  $P_{pc}$  detected by the motor pressure sensors 53, 54, inlet pressure to the hydraulic motor 50 is referred to as  $P_b$ . Incidentally, an output shaft rotation speed can be substituted for the motor rotation speed  $\omega_{out}$ .

[0033] In step 312, an acceleration  $V_a$  of the vehicle is calculated from the change of the vehicle speed  $V$ , that is, determined by excluding the former  $\omega_{out}$  1 from the  $\omega_{out}$  2 of this time ( $V_a = \Delta\omega_{out} = \omega_{out} 2 - \omega_{out} 1$ ).

[0034] In step 313, an absorbing torque command value  $TTVC$  of the hydraulic pump 10 is calculated. This is effected in such a manner that the absorbing torque is obtained by a map of Fig. 7 from the engine rotation speed  $\omega_e$  and the motor rotation speed  $\omega_{out}$ , and the control device 60 outputs the absorbing torque command value  $TTVC$  to the TVC valve 11a which controls discharge amount of the hydraulic pump 10.

[0035] Here, an absorbing torque control of the hydraulic pump will be described. Fig. 7 shows the horizontal axis of the engine rotation speed  $\omega_e$ , the vertical axis of the engine torque  $T_e$ , and the absorbing torque  $TTVC$  of the hydraulic pump 10 due to the TVC valve 11a. In addition, the accelerating amount  $\theta$  ( $\theta_0 = 0$ ,  $\theta_1 = 1/4$ ,  $\theta_2 = 2/4$ ,  $\theta_3 = 3/4$ ,  $\theta_4 = 4/4$  Full) is represented by slanting dotted lines. A solid line  $V_{dh}$  represents a change of the TVC absorbing torque when the vehicle speed is high, and a solid line  $V_{dl}$  represents a change of the same when the vehicle speed is low.

[0036] Referring to Fig. 2, upon receipt of the vehicle speed  $V$  from the vehicle speed sensor 52, and the engine rotation speed  $\omega_e$  from the engine rotation speed sensor 3, the control device 60 outputs a command to the TVC valve 11a so that the pump discharge amount  $x$  discharge pressure changes as predetermined in response to the engine rotation speed  $\omega_e$ .

[0037] For example, when the position of the shifter 63 is selected to neutral  $N$  at the position where the accelerating amount  $\theta$  of  $\theta_0 = 0$ , and at the position where the engine rotation speed  $\omega_e$  is on the point  $X_1$ , the vehicle is suspending, or moving at slow speed by dragging torque. At this time, the hydraulic pump 10 discharges a predetermined minimum capacity, and discharged oil returns to the oil tank 6 from the unload valve 13.

[0038] When the shifter 63 is selected to forward  $F$ , the control device 60 sends a signal responsive to the accelerating amount to the forward solenoid operated directional control valve 41 by the signal from the shifter selected position sensor 63a. The solenoid operated directional control valve 41 is switched by the signal in response to the accelerating amount, controls the pilot

pressure to send to the running valve 21, and opens the running valve 21 at a predetermined opening degree. When the accelerator pedal 61 is further depressed, the running valve 21 maintains a predetermined constant opening degree until a position where the accelerating amount  $\theta$  is near the front of  $\theta_1 = 1/4$  (the engine rotation speed  $\omega_e$  is on the position of the point  $X_2$ ). Therefore, until the point  $X_2$ , the vehicle is moving at slow speed by dragging torque in the same manner as described above, so that garaging, etc. are facilitated.

[0039] Further, when the accelerator pedal 61 is depressed, and the accelerating amount  $\theta$  is on the position near the after of  $\theta_1 = 1/4$  (the engine rotation speed  $\omega_e$  is on the position of point  $X_3$ ), the running valve 21 maintains a full opening. At the same time, the control device 60 sends a command to the TVC valve 11a to actuate the servo device 11, and increases the absorbing torque (for example, displacement or pressure). This reduces a shock at the time of starting because the vehicle is started by the dragging torque at the beginning, and the speed increases after a while, and a smooth starting can be effected without any resistance because the running valve 21 is fully opened. In addition, reduction in heating value and an improvement in running efficiency can be obtained. When the engine rotation speed  $\omega_e$  passes the point  $X_3$ , displacement of the hydraulic pump 10 increases, and the absorbing torque of the hydraulic pump 10 increases rapidly, thereby responding to the change of the vehicle speed with good follow-up properties. In addition, a large engine torque  $T_e$  is taken when the vehicle speed is low for the purpose of generating a feeling such as of a torque converter.

[0040] In step 314, a fuel injection amount command value  $\omega_{ec}$  to the engine 1 is calculated. That is, the engine rotation speed  $\omega_e$  is obtained from the accelerating amount  $\theta$  and the motor rotation speed  $\omega_{out}$  shown in a map of Fig. 8, and a command signal is outputted from the control device 60 to the solenoid 2a of the fuel injection pump 2 which controls the engine rotation speed  $\omega_e$ .

[0041] Here, the control of the engine rotation speed  $\omega_e$  will be described. Referring to Fig. 8, the horizontal axis represents an accelerating amount  $\theta$ , the vertical axis represents the engine rotation speed  $\omega_e$ , and the vehicle speed  $V$  is represented by the solid lines. The solid line  $V_{dh}$  represents the change of the engine rotation speed  $\omega_e$  when the vehicle speed is high, and the solid line  $V_{dl}$  represents the same when the vehicle speed is low. When the vehicle speed is high, the gradient of the change of the engine rotation speed  $\omega_e$  is increased, and the gradient is decreased when the vehicle speed is low.

[0042] Upon receipt of the vehicle speed  $V$  from the vehicle speed sensor 52 and the accelerating amount  $\theta$  from the accelerating amount detection sensor 61a, the control device 60 outputs a command value  $\omega_{ec}$  to the solenoid 2a so that the rotation speed changes as predetermined in response to the accelerating amount  $\theta$  fol-



lowing Fig. 8. By the command vale  $\omega_{ec}$ , a predetermined injection amount is sent to the engine 1, and the engine rotation speed  $\omega_e$  becomes a predetermined rotation speed. For example, when the solid line  $V_{dh}$  of running at high speed, the maximum rotation speed is increased to absorb engine outputs as much as possible. Conversely, when low speed, motor output torque is required, so the maximum rotation speed of the engine 1 is set low. In addition, a range of an idling rotation speed is prepared, and when changing to the idling from the high speed running, the idling is set relatively high so as to increase cooling capability.

**[0043]** In step 315, a flow rate command value  $Q$  of the running valve 21 is calculated. This is effected in such a manner that a flow rate  $Q$  of the running valve 21 (an opening degree amount command value  $L$  of the running valve 21) is obtained by a map shown in Fig. 9 from the accelerating amount  $\theta$  and the motor rotation speed  $\omega_{out}$  (the vehicle speed  $V$  of the vehicle), and the flow rate command value  $Q$  signal is outputted from the control device 60 to the solenoid operated directional control valves 41, 42 which control the flow rate  $Q$  of the running valve 21.

**[0044]** Here, the control of the opening degree of the running valve 21 will be described. Referring to Fig. 9, the horizontal axis represents the vehicle speed  $V$  of the vehicle, the vertical axis represents the opening degree amount command value  $L$  of the running valve 21, and the accelerating amount  $\theta$  ( $\theta_0 = 0$ ,  $\theta_2 = 1/3$  Full,  $\theta_3 = 2/3$  Full,  $\theta_4 =$  Full) is represented by solid lines. The lower side of the slanting dotted line  $L_b$  represents a brake region LDB, and the upper side represents a running region LDD. In addition, in the running region LDD, opening of a spool shows a full opening region LDF when the opening degree command value of the running valve 21 is not less than a predetermined value  $L_a$ . Further, a portion of the slanting solid line of the accelerating amount  $\theta$  represents a speed balance region (a CLSS control region), and a control by the running valve 21 is effected in the speed balance region. In the full opening region LDF, the opening degree of the running valve 21 is provided with predetermined openings, each becoming full in accordance with the accelerating amount  $\theta$ , so that even a small accelerating amount  $\theta$  is included in the full opening region LDF. This reduces a flow rate resistance, eliminates pressure collapse, and reduces heating.

**[0045]** Incidentally, although the opening degree amount value is shown in Fig. 9, the opening degree of the running valve 21 is proportional to the flow rate ( $Q = K \cdot L$ ) in the CLSS control. Thus, the flow rate command value  $Q$  for flowing from the hydraulic pump 10 to the hydraulic motor 50 will be obtained based on the opening degree of the running valve.

**[0046]** In the brake region LDB, the opening degree amount command value  $L$  of the running valve 21 is changed in response to the accelerating amount  $\theta$ . In this range, the opening degree amount command value

$L$  of the running valve 21 is not closed, and the larger the accelerating amount  $\theta$ , the more the opening degree amount command value  $L$  is increased. Thus, even if brakes are applied to the vehicle, the vehicle does not stop suddenly, but suitably reduces speed. Further, at the left side of  $\theta_1$  shown in the drawing, similar to the above-description, a creep running is performed in which the vehicle moves at slow speed by a dragging torque. In addition, in the running mode D, the vehicle rarely runs by the control in the speed balance control between the full opening region LDF and the brake region LDB.

**[0047]** Referring to Fig. 9, upon receipt of the signals of the vehicle speed  $V$  and the accelerating amount  $\theta$ , the control device 60 reduces the running in the speed balance control, minimizes pressure collapse due to the running valve 21, and obtains the command valve  $Q$  to be outputted to the solenoid operated directional control valves 41, 42 in accordance with Fig. 9 so as to excellently maintain brake properties.

**[0048]** For example, when the vehicle is running with the accelerating amount  $\theta$  on the position of  $\theta_4 =$  Full, and at high-speed (point X5) vehicle speed  $V_c$ , the accelerating amount  $\theta$  and the vehicle speed  $V_c$  are detected by each of the sensors, and sent to the control device 60. The control device 60 receiving the signals outputs an opening degree command value  $L_c$  of the full opening region LDF for minimizing pressure collapse to the running valve 21 through the solenoid operated directional control valves 41, 42. Further, when the accelerating amount  $\theta$  remains as it is, and the vehicle speed  $V_c$  increases due to a downhill, etc., the vehicle speed  $V_c$  and the opening degree command value  $L_c$  change along the line of the accelerating amount  $\theta_4 =$  Full.

**[0049]** At this time, from point X6 to point X7 on the line of the accelerating amount  $\theta_4 =$  Full, the speed balance control (CLSS control region), i.e., the opening amount of the running valve 21 gradually reduces, and a speed control for giving a braking force is effected. When the speed increases not less than the point X7, the opening amount of the running valve 21 becomes a predetermined constant value of the line  $L_d$ , and the running valve 21 gives a predetermined resistance corresponding to the speed, and generates a braking force. Over the high speed point X6, the two-stage back pressure valve 23 actuates to control the return pressure of the return oil to the tank 6 to a high pressure, and oil is supplied to the hydraulic motor 50 from either of the hydraulic pump 10 and/or suction valve 31 and the suction valve 32.

**[0050]** In addition, when the shifter 63 for selecting the travelling direction of the vehicle is in forward F or reverse R, at the left side of the accelerating amount  $\theta$  of  $\theta_a$  shown in the drawing, the running valve 21 is opened by a predetermined small amount, thereby providing creep running. This creep running enables shock reduction and slow speed forwarding, as described above.

**[0051]** In step 316, a low-pass filter shown in Fig. 10



is multiplied by the flow rate command value Q of the running valve 21. That is, a command signal to solenoid operated directional control valves 41, 42 for switching the running valve 21 removes noises of not less than high-frequency f0 with the low-pass filter (this is expressed by  $Q_a = G_a \times Q$  in terms of a transfer function). This prevents very small pressure vibrations of the running valve 21, so that speed changes of the vehicle due to vibrations of the running valve 21 is eliminated.

[0052] Next, in step 317 shown in Fig. 5, whether or not the shifter 63 is operated, i.e., the position of the shifter 63 is judged by the shifter selected position sensor 63a. If neutral N, the procedure advances to step 318. If forward F, the procedure advances to step 319. If reverse R, the procedure advances to step 320.

[0053] In step 318 (in the case of neutral N position), it is checked that a forward flow rate command value QF from the control device 60 to the forward solenoid operated directional control valve 41, and a reverse flow rate command value QR to the solenoid operated directional control valve 42 are zero. When agreed with the judgement in step 317, the procedure advances to step 321.

[0054] In step 319 (in the case of forward F position),  $Q_a$  is substituted into the forward flow rate command value QF, and the reverse flow rate command value QR is reduced to zero. Further, pressure Pca of the forward side pipe 56 is substituted into the inlet pressure Pp so as to be used for controlling an inclined shaft of the hydraulic motor 50. When agreed with the judgement in step 317, the procedure advances to step 321.

[0055] In step 320 (in the case of reverse R position),  $Q_a$  is substituted into the reverse flow rate command value QR, and the forward flow rate command value QF is reduced to zero. Further, pressure Pcb of the reverse side pipe 57 is substituted into the inlet pressure Pp so as to be used for controlling the inclined shaft of the hydraulic motor 50 as in the case of forward F.

[0056] In step 321, whether the running region LDD or the brake region LDB is judged by a map shown in Fig. 9. If the running region LDD, the procedure advances to step 322. If the brake region LDB, the procedure advances to step 327.

[0057] In step 322, an inclined shaft command value Dd of the hydraulic motor 50 on running is calculated. The inclined shaft command value Dd controls the inclined shaft of the hydraulic motor 50 on running to control discharge capacity of the hydraulic motor 50. The motor inclined shaft control will be described with Fig. 11.

[0058] Referring to Fig. 11, the horizontal axis represents the inlet pressure Pp to the hydraulic motor 50, and the vertical axis represents the inclined shaft command value Dd of the hydraulic motor 50. The solid line Ddh shows a change of the inclined shaft command value Dd when the vehicle speed V is high, and the solid line Ddl show the same when the vehicle speed V is low. For example, when the vehicle speed is high (the solid line Ddh), the inclined shaft command value Dd is set

relatively small, and the gradient of the change is increased. In addition when the vehicle speed is low (the solid line Ddl), the inclined shaft command value Dd is set relatively large, and the gradient is decreased. In addition, until the inlet pressure Pp reaches a predetermined value Ppa, the inclined shaft command value Dd maintains the minimum inclined shaft angle, and further, after the predetermined value Ppb, the inclined shaft command value Dd maintains the maximum inclined shaft angle.

[0059] In addition, based on the inlet pressure Pp detected by the motor pressure sensors 53, 54, and the vehicle speed Va detected by the vehicle speed sensor 52, the inclined shaft command value Dd is obtained from the map. The control device 60 outputs the thus obtained inclined shaft command value as a command value to the motor solenoid operated directional control valve 55 of the servo device 51.

[0060] At the time of the outputting, for example, when running at high speed of a small required torque, the inclined shaft command value Dd is decreased and the gradient of the change of the inclined shaft command value Dd is increased, thereby improving follow-up properties with respect to torque changes. In addition, when low speed of the large required torque, the inclined shaft command value Dd is increased, and the gradient of the change of the inclined shaft command value Dd is decreased to prevent hunting of the hydraulic motor 50 with respect to the torque changes. Since the inclined shaft command value Dd suitably changes in response to the required torque by providing a selection range between the high speed and the low speed, and providing a constant vehicle speed region, the percentage of change of the vehicle speed is reduced and a good response to the change of the vehicle speed is provided, thereby improving driving properties.

[0061] In step 323, changes in acceleration are examined. This judges whether or not the acceleration Va of the vehicle speed V is larger than a predetermined acceleration Vamax (a threshold value which can be arbitrarily set). If not, i.e., if smaller, the procedure advances to step 324. If larger, the procedure advances to step 328.

[0062] When the procedure advances to step 328, since the vehicle speed is increased (descending a slope, etc.) greatly, the rotation speed of the hydraulic motor 50 changes faster than a predetermined amount. For this reason, the discharge amount from the hydraulic pump 10 cannot follow up, so the control device 60 sends a signal to the two-stage back pressure valve 23, and switches the two-stage back pressure valve 23 in step 328 in the direction in which high pressure is generated. When switched to the high pressure, the discharge side return oil from the hydraulic motor 50 is fed to the pipe 56 or the pipe 57 of the supply side from the hydraulic pump 10 through the suction valve 31 or the suction valve 32 so as to prevent occurrence of cavitation.

[0063] In step 324, entrance pressure of the hydraulic motor 50 is examined. This judges whether or not the supply side pressure  $P_d$  to the hydraulic motor 50 is lower than the threshold value of a predetermined pressure  $P_{dmin}$ . If not, i.e., higher than the threshold value, the procedure advances to step 325 because a non-illustrated vehicle is driven by the hydraulic motor 50. On the other hand, when lower than the threshold value, the procedure advances to step 328, and the two-stage back pressure valve 23 is switched in the direction in which high pressure is generated in the same manner as described above.

[0064] In step 325, the percentage of change of the accelerating amount  $\theta$  is detected. An amount of change  $\theta_a$  of the accelerating amount  $\theta$  is obtained by the difference between the former accelerating amount  $\theta_f$  and accelerating amount  $\theta_n$  of this time, and whether or not the difference  $\theta_a$  is larger than a predetermined threshold value  $\theta_{dec}$  is judged. If not larger, a change of commanded speed is considered to be small, and the procedure advances to step 326. If larger, the procedure advanced to step 328.

[0065] In step 326, the two-stage back pressure valve 23 is turned off, i.e., the two-stage back pressure valve 23 is controlled to a low pressure by the command from the control device 60, whereby return oil is mainly returned to the oil tank 6. In addition, a part of return is fed to either of the supply side pipe 56 and 57 as needed through the suction valve 31 or the suction valve 32. When step 326 is completed, the procedure advances to step 329.

[0066] In step 327 to which the procedure advances in the case of the brake region LDB in the above step 321, the inclined shaft command value  $D_d$  of the hydraulic motor 50 on braking is calculated. The control of the discharge capacity of the hydraulic motor 50 on braking will be described with Fig. 12. Dotted line  $Dbh$  represents the change of the inclined shaft command value  $D_d$  when a brake depressing angle of the brake pedal 62 is high, and the solid line  $Dbi$  represents the same when the brake pedal 62 is not depressed. When the vehicle speed is high, the inclined shaft command value  $D_d$  is decreased, and the inclined shaft command value  $D_d$  is increased when the vehicle speed is low. By this, when the vehicle speed  $V$  is low, the inclined shaft command value  $D_d$  is increased, whereby a braking torque amount is increased to apply hard braking and improve follow-up properties. When the vehicle speed  $V$  is high, braking is slowed down to prevent the hard braking and maintain safety. When the discharge capacity control of the hydraulic motor 50 on braking is completed, the procedure advances to step 329 via step 328.

[0067] In step 329, a low-pass filter shown in Fig. 13 is multiplied by the inclined shaft command value  $D_d$  on braking. This allows a command signal to the solenoid operated directional control valve 55, which generates pressure for switching the hydraulic motor to pass through the low-pass filter, so as to remove high-frequency noises.

This is represented by  $D_{dc} = G_b \times D_d$  in terms of a transfer function. By the low-pass filter, very small vibrations of the hydraulic motor 50 due to high frequency is prevented, thereby eliminating speed changes of the vehicle.

[0068] In step 330, the forward flow rate command value  $Q_f$  is converted into a current. In addition, in step 331, the reverse flow rate command value  $Q_r$  is converted into a current. Referring to Fig. 14, the horizontal axis represents the forward flow rate command value  $Q_f$ , the vertical axis represents the current  $I_f$  to the forward solenoid operated directional control valve 41. Referring to Fig. 15, the horizontal axis represents the reverse flow rate command value  $Q_r$ , and the vertical axis represents the current  $I_r$  to the reverse solenoid operated directional control valve 42. Incidentally, in the vertical axes of Fig. 14 and later drawings, the current to the solenoid operated directional control valve is referred to as "EPC current".

[0069] As shown in Fig. 14 and Fig. 15, the forward current  $I_f$  or the reverse current  $I_r$  is changed with respect to the forward flow rate command value  $Q_f$  or the reverse flow rate command value  $Q_r$ , and variations of the running valve 21 and the solenoid operated directional control valves 41, 42 are absorbed. The map is set so that the discharge capacity of the hydraulic motor 50 varies during forwarding and during reversing, and the discharge capacity during reversing is large. This increase the inclined shaft command value  $D_d$  on reversing larger than that of during forwarding to provide a large output torque, so that escape from uneven ground, etc. is facilitated.

[0070] In step 332, a current conversion command value (current  $I_{dm}$ ) is obtained by a map shown in Fig. 16 from the inclined shaft command value  $D_d$  passing through the low-pass filter, and outputted to a solenoid operated directional control valve 55. The horizontal axis represents the inclined shaft command value  $D_d$ , and the vertical axis represents the current  $I_{dm}$  to the solenoid operated directional control valve 55. The current  $I_{dm}$  is changed as in the case of Fig. 14 and Fig. 15 to correct non-linearity of the hydraulic motor 50 and the solenoid operated directional control valve 55.

[0071] In step 333, the command signal to the TVC valve 11a obtained in step 313 is converted into a current  $I_{TVC}$  by a map shown in Fig. 17 to output a command. The horizontal axis represents the absorbing torque command value  $T_{TVC}$  to the TVC valve 11a, and the vertical axis represents the current  $I_{TVC}$  to the TVC valve 11a.

[0072] In step 334, the command signal  $\omega_{ec}$  to the solenoid 2a of the fuel injection pump 2 obtained in step 314 is converted into a current  $I_e$  by a map of Fig. 18 to output a command. The horizontal axis represents the rotation speed command value  $\omega_{ec}$  of engine 1, and the vertical axis represents the current  $I_e$  to the solenoid 2a. When step 334 is completed, the procedure returns to step 311. By the flow as described above, running and

brake control in the running mode D are effected.

[0073] Next, a description will be given of the usage of the vehicle by an operator in the running mode D. When only running for transportation, etc. is required, the operator selects the running mode D by the mode selection switch 64. Then, the engine 1 is started. However, in a situation where the accelerator pedal 61 is not depressed, the engine 1 is on the position of idling and rotates at low speed, as shown in Fig. 8. When the operator operates the shifter 63 for selecting the vehicle travelling direction, and depresses the accelerator pedal 61, the vehicle will run. At this time, the control device 60 sets the rotation speed  $\omega_e$  of the engine 1 by Fig. 8, the running hydraulic pump 10 by Fig. 7, the opening degree amount of the running valve 21 by Fig. 9, and the inclined shaft command value  $D_d$  by Fig. 11, respectively, based on the signals from the shifter selected position sensor 63a and the accelerating amount detection sensor 61a, whereby the vehicle runs at a predetermined running speed.

[0074] By the selection of the above-described running mode D, the rotation speed  $\omega_e$  of the engine 1 is controlled by a depressing amount of the accelerator pedal 61. In addition, at the time of a normal running, the opening degree amount of the running valve 21 is on the position where the opening of the spool is in the full opening region LDF, and resistance at the time of running is reduced. Further, at the time of running, if the speed is increased higher than that corresponding to the accelerating amount, the opening degree amount of the running valve 21 enters the speed balance region LDD where the opening of the spool is restricted, so that the speed is controlled by the running valve 21. However, the period of time is very short, and the opening region LDF immediately enters the brake region LDB and is controlled. When entered the brake region LDB, the two-stage back pressure valve 23 is operated following the steps of the above flow, thereby preventing occurrence of cavitation.

[0075] At the time of brake operation, when the operator depresses the brake pedal 62, a signal from the braking amount detection sensor 62 enters the control device 60, and the control device 60 controls the discharge capacity of the hydraulic motor 50 in response to the depressing amount of the brake pedal 62 shown in Fig. 12 and the vehicle speed V, and applies a brake.

[0076] Next, a case will be described where the operation mode W is selected in operating mode selection.

[0077] First, an operation mode will be described. At the time of operation, if the mode selection switch 64 is switched to select the operation mode W, the operation mode W is detected by the mode detection sensor 64, and a signal of the operation mode W is sent to the control device 60. In the control device 60, a map of the operation mode W is accessed and judged. For example, only the operating machine lever is operated, the operating machine CLSS valves 46a, 46b, ... are actuated, and the flow rate of each CLSS valve and the dis-

charge amount of the hydraulic pump 10 are controlled by the highest pressure PS in the CLSS valves. In addition, when the operating machine lever is operated while running, the discharge amount of the hydraulic pump 10, and the flow rate of the operating machine CLSS valve 46a and the running valve 21 are controlled by a higher pressure which acts on the operating machine CLSS valve 46a and the running valve 21.

[0078] The hydraulic driving control when the operation mode W is selected, i.e., a case where the procedure advances to step 303 in the flow of Fig. 3 will be described with the flows of Fig. 19 to Fig. 21.

[0079] Referring to Fig. 19, in step 351, various signals are inputted to the control device 60. That is, the control device 60 reads in ever-changing signals from the various sensors. The signals, as the similar signals in step 311, include a change of the position of the shifter 63 (changes of forward F, reverse R, neutral N), the engine rotation speed  $\omega_e$ , the accelerating amount  $\theta$ , the rotation speed  $G_{out}$  of the hydraulic motor 50 (the vehicle speed V), the inlet and outlet pressures  $P_{pc}$ , and the braking capacity  $R_c$  of the retarder. In this step, an engine rotation setting dial Erev for setting rotation of the engine 1, an operation mode selecting signal Sv for setting outputs of the operating machine, a heavy digging mode, a digging mode, and a fine operation mode, etc. according to works are further inputted.

[0080] In step 352, similar to step 312, an acceleration  $V_{sa}$  of the vehicle is calculated from the change of the vehicle speed V in the operation mode W. This calculation is made by excluding the former  $\omega_{out 1}$  from the  $\omega_{out 2}$  of this time, and is represented by the following expression:

$$V_{sa} = \Delta \omega_{out} = \omega_{out 2} - \omega_{out 1}$$

[0081] In step 353, an absorbing torque TSTVC of the hydraulic pump 10 in the operation mode W is calculated. This is effected in such a manner that the absorbing torque of the hydraulic pump 10 is obtained from the engine rotation speed  $\omega_e$ , the engine rotation setting dial Erev and the operation mode selecting signal Sv based on a map shown in Fig. 22. The obtained absorbing torque TSTVC is outputted as a command signal from the control device 60 to the TVC valve 11a.

[0082] The absorbing torque control of the hydraulic pump 10 in this case will be described. Referring to Fig. 22, the horizontal axis, the vertical axis, and the accelerating amount of dotted lines are same as those of Fig. 7. Solid lines of hyperbolas represent the absorbing torque in the operation mode W from the operation mode selecting signal Sv, high absorbing torque in a heavy cutting mode is represented by a solid line TSH, the change of the absorbing torque of the TVC when the absorbing torque is low in the fine operation mode is represented by a solid line TSL. Upon receipt of the vehicle speed V of the vehicle from the vehicle speed sensor

52, the engine rotation speed  $\omega_e$  from the engine rotation speed sensor 3, and the operation mode selecting signal Sv, the control device 60 outputs a command to the TVC valve 11a so that the pump discharge amount x discharge pressure changes as predetermined in response to the engine rotation speed  $\omega_e$ .

[0083] For example, when the operator selects the heavy cutting mode by the mode selection switch 64, the operation mode selecting signal Sv of the heavy cutting mode is sent to the control device 60 from the mode detection sensor 64a, whereby the control device 60 selects the solid line TSH of high absorbing torque from a non-illustrated storage device. At this time, when the operator sets the engine rotation setting dial Erev to the position of the accelerating amount  $\theta_4 = \text{Full}$ , the control device 60 selects the dotted and slanting line of the accelerating amount  $\theta_4 = \text{Full}$  from the storage device. Sequentially, when a non-illustrated operating machine lever is operated, an operating machine solenoid operated directional control valve (not shown) is actuated upon receipt of a signal from the control device 60 in response to an operating amount so as to switch the operating machine CLSS valve 46a.

[0084] In accordance with the actuation, pressure from the TVC valve 11a is sent to the servo 11c so that the LS differential pressure ( $PLS = PP - PS$ ) is held constant, and the inclination-rotation angle (discharge amount of the pump) of the hydraulic pump 10 is controlled. When the operating amount is increased and the discharge amount of the pump is increased, or when pressure applied to the hydraulic pump 10 is increased due to load increase of the operating machine, the absorbing torque TTVC of the hydraulic pump 10 reaches a torque curve TSH, and varies along the torque curves TSHa, TSHb, TSHc, and TSHd. That is, the absorbing torque TTVC moves sequentially on the solid lines along the points (TSHa, TSHb, TSHc, TSHd) in response to the load and the engine rotation speed  $\omega_e$ .

[0085] In contrast, when the operator sets the engine rotation setting dial Erev to the position of the accelerating amount  $\theta_3 = 3/4$  by the mode selection switch 64, and selects the fine operation mode, the absorbing torque TTVC sequentially moves on the solid lines along the points (TSLa, TSLb, TSLc, TSLd) in response to the load and the engine rotation speed  $\omega_e$ . This reduces the absorbing torque TTVC of the hydraulic pump 10 to be smaller than the absorbing torque in the heavy cutting mode, and requires less engine torque, so that reduction in fuel economy can be achieved.

[0086] In addition, as the hydraulic pump 10, one variable displacement hydraulic pump is used. The hydraulic pump 10 sends pressure oil to actuators for driving the hydraulic driving apparatus for vehicles and an operating machine so as to drive each of them, and controls them so that the discharge amount falls within the predetermined range in response to the rotation speed of the hydraulic motor 50 and the engine. By this, in the vehicle having the operating machine, one hydraulic

pump 10 is used for both operating machine and running, whereby space and cost of the vehicle can be reduced.

[0087] In step 354, a fuel injection amount command value of the engine 1 in the operation mode W is calculated. Here, the control device 60 obtains the engine rotation speed  $\omega_e$  from the engine rotation setting dial Erev by a map shown in Fig. 23, and outputs a command signal to the solenoid 2a which controls the engine rotation speed  $\omega_e$ . Referring to Fig. 23, a solid line Trev (fuel injection amount command value Trev) represents the lineal change of the engine rotation speed from the idle rotation speed to the maximum rotation speed.

[0088] For example, when the operator wishes to operate the operating machine quickly for the heavy cutting, the engine rotation setting dial Erev is set to the maximum rotation speed (full-throttle position). On the other hand, when the operator wishes to operate the operating machine slowly in the fine operation, the engine rotation setting dial Erev is set near the idle rotation speed. This allows the engine 1 to rotate corresponding to the operating amount of the engine rotation setting dial Erev. Incidentally, according to this embodiment, the control of the engine rotation speed  $\omega_e$  during the operation mode W, i.e., the fuel injection amount command value of the engine is set by the operating amount of the engine rotation setting dial Erev rather than the operating amount of the accelerator pedal 62.

[0089] In step 355, correction of the accelerating amount  $\theta$  is obtained. This is effected in such a manner that an accelerating correction amount  $\theta_s$  is obtained from the accelerating amount  $\theta$  by Fig. 24, and a command signal is outputted from the control device 60 to the solenoid 2a of the fuel injection pump 2 which controls the engine rotation speed  $\omega_e$ . Fig. 24 shows the relationship between the accelerating amount  $\theta$  and the accelerating correction amount  $\theta_s$  in each engine rotation speed  $\omega_e$ , and a solid line NSHA represents the maximum rotation speed of the engine 1, and a solid line NSHB represents the idle rotation speed. In this embodiment, the accelerating correction amount  $\theta_s$  is changed corresponding to the engine rotation speed  $\omega_e$  with respect to the same accelerating amount  $\theta$ , and the accelerating correction amount  $\theta_s$  at the idle rotation speed is smaller than the maximum rotation speed.

[0090] The correction will be described. Since the flow rate is determined by the opening amount of the running valve (closed center load sensing valve) 21 regardless of the discharge amount of the hydraulic pump 10, the opening amount is fixed and the flow rate is also fixed in the same accelerating amount  $\theta$ . For this reason, if the opening amount for forwarding the vehicle at slow speed is set, the opening amount cannot be increased and the running speed is slow when a normal speed running is required. To solve this, according to this embodiment, the opening amount is increased to the normal speed, and the opening amount can be decreased when a slow speed forwarding is required. Therefore, the ac-

celerating correction amount  $\theta_s$  corrected with respect to the accelerating amount  $\theta$  is set so as to output a small command.

[0091] In step 356, the control device 60 calculates the flow rate command value  $Q_s$  of the running valve 21 (the opening degree command value  $L_s$  of the running valve 21) during the operation mode W by a map shown in Fig. 25 from the accelerating amount  $\theta$  and the motor rotation speed  $\omega_{out}$  (the vehicle speed  $V$  of the vehicle), and outputs a command signal to the solenoid operated directional control valves 41, 42 for controlling the flow rate  $Q_s$ . Referring to Fig. 25 which illustrates the opening degree control of the running valve 21, the horizontal axis represents the vehicle speed  $V$  of the vehicle, and the vertical axis represents the opening degree command value  $L_s$  (the flow rate command value  $Q_s$ ) of the running valve 21. In addition, a solid line substantially parallel to the horizontal axis represents the accelerating correction amount  $\theta_s$  ( $\theta_{s0} = 0$ ,  $\theta_{s1} = 1/4$  Full,  $\theta_{s2} = 2/4$  Full,  $\theta_{s3} = 3/4$  Full,  $\theta_{s4} =$  Full). The upper side of the dotted line  $L_{\theta sa}$  represents the running region LDD, the lower side of the dotted line  $L_{\theta sb}$  represents the brake region LDB, and a portion between the dotted line  $L_{\theta sa}$  and the dotted line  $L_{\theta sb}$  represents the speed balance region, respectively.

[0092] Referring to Fig. 25, upon receipt of the vehicle speed  $V$  from the vehicle speed sensor 52 and the accelerating amount  $\theta$  from the accelerating amount detection sensor 61a, the control device 60 obtains the accelerating correction amount  $\theta_s$  responsive to the accelerating amount  $\theta$ . And, until the dotted line  $L_{\theta sa}$  where the vehicle speed reaches the predetermined vehicle speed  $V$ , the control device 60 performs the above-described CLSS control (the flow rate control responsive to the opening degree amount of the running valve 21) in response to the accelerating correction amount  $\theta_s$ . In addition, at the predetermined vehicle speed  $V$  or more, the control device 60 decreases the opening degree command value  $L_s$  to the running valve 21 even if each of the accelerating correction amounts  $\theta_s$  are equal. Therefore, at the vehicle speed above the dotted line  $L_{\theta sa}$ , the speed balance region is provided. In addition, even if each of the accelerating correction amounts  $\theta_s$  are equal, the brake region LDB is provided at the vehicle speed above the dotted line  $L_{\theta sb}$ , and the brake region LDB is changed gradually in accordance with the accelerating correction amount  $\theta_s$ .

[0093] In the above-described running region LDD, the control device 60 outputs the opening degree command value  $L_s$  from the corrected accelerating correction amount  $\theta_s$  and the vehicle speed  $V$  to the solenoid operated directional control valve 41 or 42, and controls to effect the CLSS control. In the brake region LDB, the control device 60 changes the opening degree command value  $L_s$  ( $=$  flow rate command value  $Q_s = K_s \cdot L_s$ ) in accordance with the corrected accelerating correction amount  $\theta_s$ . In this range, the opening degree command value  $L_s$  is not closed, and the higher the

speed, the more the opening degree command value  $L_s$  is increased. Thus, even if the brake is applied, the vehicle does not stop suddenly, but suitably reduces speed.

[0094] In the speed balance region, the control device 60, similar to the running region LDD, outputs the opening degree command value  $L_s$ , and controls so that the opening amount of the running valve 21 is decreased. At this control, the control device 60 outputs a command to the solenoid operated directional control valve 55 of the hydraulic motor 50 along from the parallel solid line of the accelerating correction amount  $\theta_s$  to the solid line off to the lower left. By the command, the inclined shaft command value  $D_{sm}$  of the hydraulic motor 50 is varied to change the capacity of the hydraulic motor 50. Further, at  $\theta_{s0}$  or less, similar to the above-description, a creep running is performed.

[0095] The above-described operation is, for example, performed in such a manner that the operator sets the engine rotation speed  $\omega_e$  to  $\theta = 3/4$  Full by the engine rotation setting dial Erev, and operates the operating lever of the operating machine and the accelerator pedal 61 to effect a simultaneous operation of the operation and the running. At the position where the accelerating correction amount  $\theta_s$  is on the position of point  $L_{s1}$  of the  $\theta_{s3}$  and the vehicle speed  $V = 0$  with respect to the accelerating amount  $\theta$ , the operating machine CLSS control is performed. The CLSS control is performed such that the operating machine CLSS valves 46a, 46b, ... are actuated by a non-illustrated operating lever, and the flow rates of each of the operating machine CLSS valves are controlled by the highest pressure  $P_S$  in the CLSS valves. At this time, a discharge flow rate  $Q_{sn}$  of the hydraulic pump 10 is the total of the operating amounts of the operating lever.

[0096] A section between the position exceeding the vehicle speed  $V = 0$  and point X11 of the dotted line  $L_{\theta sa}$ , the operating machine is operated while running, and the flow rates of the operating machine CLSS valve 46a and the running valve 21 are controlled by the higher pressure. That is, the flow rate command value  $L_{s1}$  responsive to the vehicle speed  $V$  and  $\theta_{s3}$  of the accelerating correction amount  $\theta_s$  is outputted to the running valve 21. The discharge flow rate of the hydraulic pump 10 at this time is the total flow rate  $Q_{sn}$  ( $Q_{sn} = Q_{sa} + Q_{sb} + Q_{sc} + Q_{sc}, \dots$ ) of the flow rate  $Q_{sa}$  responsive to the accelerating correction amount  $\theta_s$  and the vehicle speed  $V$ , and the flow rates ( $Q_{sb}$ ,  $Q_{sc}$ , ...) responsive to the operating amounts of each of the operating levers.

[0097] A section between point X11 of the dotted line  $L_{\theta sa}$  and point X12 of the dotted line  $L_{\theta sb}$ , the operating machine is also operated while running, but this section is the speed balance region where the discharge amount from the hydraulic pump 10 is insufficient for performing simultaneous operation. In this speed balance region, supply to the operating machine takes priority. For this reason, the flow rates consistent with the opening amount are flown to the operating machine

CLSS valves 46a, 46b,..., but the running valve 21 of the hydraulic motor 50 enters the speed balance region.

[0098] Here, the control device 60 outputs a command signal to the solenoid operated directional control valve 41 or 42 from the accelerating correction amount  $\theta_s$  and the vehicle speed  $V$ , and controls the opening degree command value  $L_s$  on line  $\theta_{s3}$  from a parallel portion of the solid line diagonally to the lower right. In addition, the control device 60 outputs a command to the solenoid operated directional control valve 55 of the hydraulic motor 50 on line  $\theta_{s3}$  from the parallel portion of the solid line diagonally to the lower right, and varies the inclined shaft command value  $D_{sm}$  to change the capacity of the hydraulic motor 50. These allow the operating machine to operate at speed consistent with the operating amount, and allow the running speed to be lower than the operating amount, whereby the simultaneous operation is effected. The brake region LDB above point X12 is similar to that described above.

[0099] In step 357, a low-pass filter of Fig. 26 is multiplied by the opening degree command value  $L_s$  (the flow rate command value  $Q_s$ ) of the running valve 21 during the operation mode W, whereby command signals to the solenoid operated directional control valves 41, 42 are removed therefrom high-frequency noises of not less than  $f_0$  (transfer function:  $Q_{sa} = G_{sa} \times Q_s$ ). The vertical axis represents a gain  $G_{sa}$ . This eliminates the speed changes of the vehicle due to vibrations of the running valve 21, similar to step 316.

[0100] In step 358 shown in Fig. 20, the position of the shifter 63 is judged. When neutral N, the procedure advances to step 359. When forward F, the procedure advances to step 360. When reverse R, the procedure advances to step 361. In the case of neutral N position, it is checked that a forward flow rate command value  $Q_{SF}$  to the forward solenoid operated directional control valve 41, and a reverse flow rate command value  $Q_{SR}$  to the reverse solenoid operated directional control valve 42 are zero. When agreed with the judgement in step 358, the procedure advances to step 362.

[0101] In step 360,  $Q_{sa}$  is substituted into the forward flow rate command value  $Q_{SF}$  to the solenoid operated directional control valve 41, and the reverse flow rate command value  $Q_{SR}$  to the solenoid operated directional control valve 42 is reduced to zero. Further, the pressure  $P_{ca}$  of the forward side pipe 56 is substituted into the inlet pressure  $P_p$ , and the procedure advances to step 362. In addition, in step 361,  $Q_{sa}$  is substituted into the reverse flow rate command value  $Q_{SR}$  to the solenoid operated directional control valve 41, and the forward flow rate command value  $Q_{SF}$  to the solenoid operated directional control valve 41 is reduced to zero. Further, the pressure  $P_{cb}$  of the reverse side pipe 57 is substituted into the inlet pressure  $P_p$ , and the procedure advances to step 362.

[0102] In step 362, whether the running region LDD or the brake region LDB is judged. That is, in Fig. 25, if the accelerating correction amount  $\theta_s$  and the rotation

speed  $\omega_{out}$  of the hydraulic motor are above the dotted line  $L\theta_{sa}$ , it is the running region LDD, while it is the brake region LDB if they are below the dotted line  $L\theta_{sb}$ . When the running region LDD, the procedure advances to step 363 to calculate the inclined shaft command value  $D_{sm}$ , which controls the discharge capacity of the hydraulic motor 50 during running in the operation mode W, by a map of Fig. 27. This control is similar to that of step 322.

[0103] In step 364, similar to step 323, it is judged whether or not the acceleration  $V_{sa}$  of the vehicle speed  $V$  is larger than the predetermined acceleration  $V_{amax}$  (threshold value). In the next step 365, similar to step 324, it is judged whether or not the supply side pressure  $P_d$  to the hydraulic motor 50 during the operation mode W is lower than the predetermined pressure  $P_{dmin}$ .

[0104] In step 366, the amount of change  $\theta_{sa}$  (the difference between the former accelerating amount  $\theta_{sf}$  and the accelerating amount  $\theta_{sn}$  of this time) of the accelerating correction amount  $\theta_s$  during the operation mode W is obtained, and whether or not the amount of change  $\theta_{sa}$  is larger than a predetermined threshold value  $\theta_{dec}$  is judged. If larger, the procedure advances to step 369 to switch the two-stage back pressure valve 23, similar to step 328. If smaller, the procedure advances to step 367 to turn off the two-stage back pressure valve 23, and then the procedure advances to step 370. In step 370, a low-pass filter shown in Fig. 29 is multiplied by the inclined shaft command value  $D_{sm}$  of the hydraulic motor 50 during the operation mode W (transfer function:  $D_{sc} = G_{sb} \times D_{sm}$ ).

[0105] When the brake region LDB is judged in step 362, the procedure advances to step 368 to calculate the inclined shaft command value  $D_{sm}$ , which controls the discharge capacity of the hydraulic motor 50 during braking in the operation mode W, by Fig. 28. After calculating, the procedure advances to step 369 to switch the two-stage back pressure valve 23, and then the procedure advances to step 370.

[0106] Step 371 to step 375 during the operation mode W are effected in the same manner as step 330 to step 334 during the running mode D. That is, in step 371, the forward flow rate command value  $Q_{SF}$  is converted into a current  $I_{sf}$  by a map of Fig. 30, and in step 372, the reverse flow rate command value  $Q_{SR}$  is converted into a current  $I_{sr}$  by a map of Fig. 31. In step 373, the inclined shaft command value  $D_{sm}$  after the low-pass filter is converted into a current  $I_{sm}$  by a map of Fig. 32, and in step 374, the pump absorbing torque command value  $T_{STVC}$  (step 353) is converted into a current  $I_{STVC}$  by a map of Fig. 33. In step 375, the command signal  $T_{rev}$  (the fuel injection amount command value  $T_{rev}$ , step 354) to the solenoid 2a is converted into a current  $I_{se}$  from a map of Fig. 34, and when step 375 is completed, the procedure returns to step 351.

[0107] The operation when the vehicle is used in the operation mode W will be described. When performing operation of loading or transporting cargo on board, and/



or running, the operator selects the operation mode W by the mode selection switch 64. Next, when the engine 1 is actuated, and the rotation speed is set by the engine rotation setting dial 66, the engine 1 starts a constant rotation. When the vehicle is operated on a fixed position, each of actuators are controlled at the highest pressure supplied to each of the actuators by the operating lever for the operating machine, and the amount of pressure oil supplied to each of the actuators is determined by the operating amount of the operation lever for the operating machine. Further, as the discharge amount of the hydraulic pump, the total oil amount of operating amount of each of the operating levers is discharged.

[0108] When performing running such as transportation while hoisting the operating machine, the engine 1 is the set constant rotation. At this time, when the shifter 63 for selecting the vehicle travelling direction is operated and the accelerator pedal 61 is depressed, the vehicle will run. However, the engine 1 may rotate at high speed and at middle speed. For this reason, when the rotation speed of the engine 1 is low, correction of an accelerator shown in Fig. 24 is effected so that the vehicle speed responsive to the accelerating amount becomes lower than the high rotation speed of the engine 1. At this time, the engine 1 is rotating at initially set rotation speed regardless of the accelerator pedal 61. In addition, when the discharge amount of the hydraulic pump 10 is insufficient, supply to an operating machine circuit takes priority. The discharge amount from the hydraulic pump 10 is controlled by the highest pressure in the CLSS valves for each of the operating machines and the running valve 21. In addition, the amount of pressure oil supplied to each of the actuators, and the running valve 21 are controlled as in the case when the vehicle is operated on the fixed position. The discharge amount of the hydraulic pump is, similar to the fixed position operation, the total oil amount of operating amounts of each of the operating levers for the operating machine. The operation during operating the brake is similar to that of during the running mode, but different in that a variable of the accelerating amount in the running mode is replaced with the accelerating correction amount.

[0109] Next, a second embodiment of the present invention will be described. The second embodiment differs from the first embodiment in the following point. That is, according to Fig. 2 of the first embodiment, the solenoid operated directional control valve 55 for the hydraulic motor 50 is switched by the command from the control device 60, and pressure of the pilot pump acts on the servo device 51 by this switching, and the servo device 51 controls the swash plate, etc. to allow displacement of the hydraulic motor 50 to be variable. In contrast, according to Fig. 35 showing the second embodiment, oil pressure from pipes 151, 152 of inlet and outlet ports of the hydraulic motor 50 is supplied to the servo device 51 via check valves 153, 154, and a solenoid operated directional control valve 155. This supply controls the swash plate, etc., which allows displacement

of the hydraulic motor 50 to be variable, similar to the first embodiment. The solenoid operated directional control valve 155 is, similarly, switched by the command from the control device 60.

[0110] Next, an operation will be described. During a normal operation, the solenoid operated directional control valve 155 is switched to a position 155a to supply oil pressure from the pipes 151, 152 of the inlet and outlet ports of the hydraulic motor 50 to the servo device 51. By this, when the pressure acting on the hydraulic motor 50 is high, pressure to be applied to the servo device 51 is increased, and the inclined shaft angle of the hydraulic motor 50 is enlarged, thereby outputting high torque. On the other hand, when the pressure acting on the hydraulic motor 50 is low, the pressure applied to the servo device 51 is decreased and the inclined shaft angle of the hydraulic motor 50 is reduced, thereby outputting high rotation speed.

[0111] In addition, when the brake is operated, the control device 60 detects that the vehicle is in the brake region LDB shown in Fig. 9 or Fig. 25, and outputs a command to the solenoid operated directional control valve 155. This command switches the solenoid operated directional control valve 155 to a position 155b, whereby the servo device 51 is connected to the tank 6. Even if the pressure acting on the hydraulic motor 50 changes, this offers a predetermined constant pressure of the servo device 50 without any changes, so that the operation of the servo device 51, which controls the swash plate, etc. is shut down. For this reason, a braking force is fixed, and the operation is facilitated.

[0112] Then, a third embodiment of the present invention will be described with Fig. 36.

[0113] This embodiment differs from the second embodiment of Fig. 35 in the following points. That is, a swivel joint 161 is provided between the running valve 21 and the hydraulic motor 50. In addition, to the return circuit from the running valve 21 to the tank 6, a pipe 162 branched between the running valve 21 and a two-stage back pressure valve 13 is connected. The pipe 162 is connected to suction valves 131, 132 and safety valves 133, 134 via the swivel joint 161.

[0114] According to such a construction, working fluid from the safety valves 133, 134 is returned to the suction valves 131, 132 at the upstream side (the side of the suction valves) of the swivel joint 161. This eliminates occurrence of cavitation. In addition, the number of pipes in the vehicle including an upper turning body can be reduced, so that the diameter of the pipe can be enlarged, and cavitation can be effectively prevented.

#### INDUSTRIAL APPLICABILITY

[0115] The present invention is useful as a speed change control method of a hydraulic driving apparatus for vehicles and a speed changing device which provides good follow-up properties with respect to the engine and the torque amount change, and which can pre-



vent hunting during low vehicle speed and provide good operability and running efficiency.

## Claims

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1. A speed change control method of a hydraulic driving apparatus for vehicles which operates a shifter to select forward and reverse of a vehicle, changes rotation speed of an engine by an accelerating amount, and supplies pressure oil from a hydraulic pump driven by the engine to a hydraulic motor through a directional control valve so as to control rotation speed of the hydraulic motor for running the vehicle wherein said method comprising:
  - discriminating between at least a power running and a brake running from said accelerating amount ( $\theta$ ) and the rotation speed ( $\omega_{out}$ ) of said hydraulic motor (50); and allowing return oil from said hydraulic motor (50) to a tank (6) to have a high pressure so as to be fed to the supply side of said hydraulic motor (50) if inlet pressure ( $P_p$ ) to said hydraulic motor (50) is lower than permissible suction pressure of said hydraulic motor (50) when discriminating the brake running.
2. A speed changing device of a hydraulic driving apparatus for vehicles including an engine of which rotation speed changes in response to an accelerating amount, a hydraulic pump driven by the engine, a brake pedal, and a hydraulic motor for receiving pressure oil from the hydraulic pump through a directional control valve to run a vehicle, said device comprising:
  - motor pressure sensors (53, 54) for detecting inlet pressure ( $P_p$ ) to said hydraulic motor (50) and outlet pressure from said hydraulic motor (50); a braking amount detection sensor (62a) for detecting a braking amount of said brake pedal (62); a variable pressure two-stage back pressure valve (23) for controlling return pressure of a return circuit (22) formed between said directional control valve (21) and an oil tank (6); and a control device (60), wherein said control device (60), at the time of the braking, compares said detected inlet pressure ( $P_p$ ) with permissible suction pressure of said hydraulic motor (50), and outputs a command to said two-stage back pressure valve (23) to increase said return pressure when said detected inlet pressure ( $P_p$ ) is lower than said permissible suction pressure.

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FIG. 1

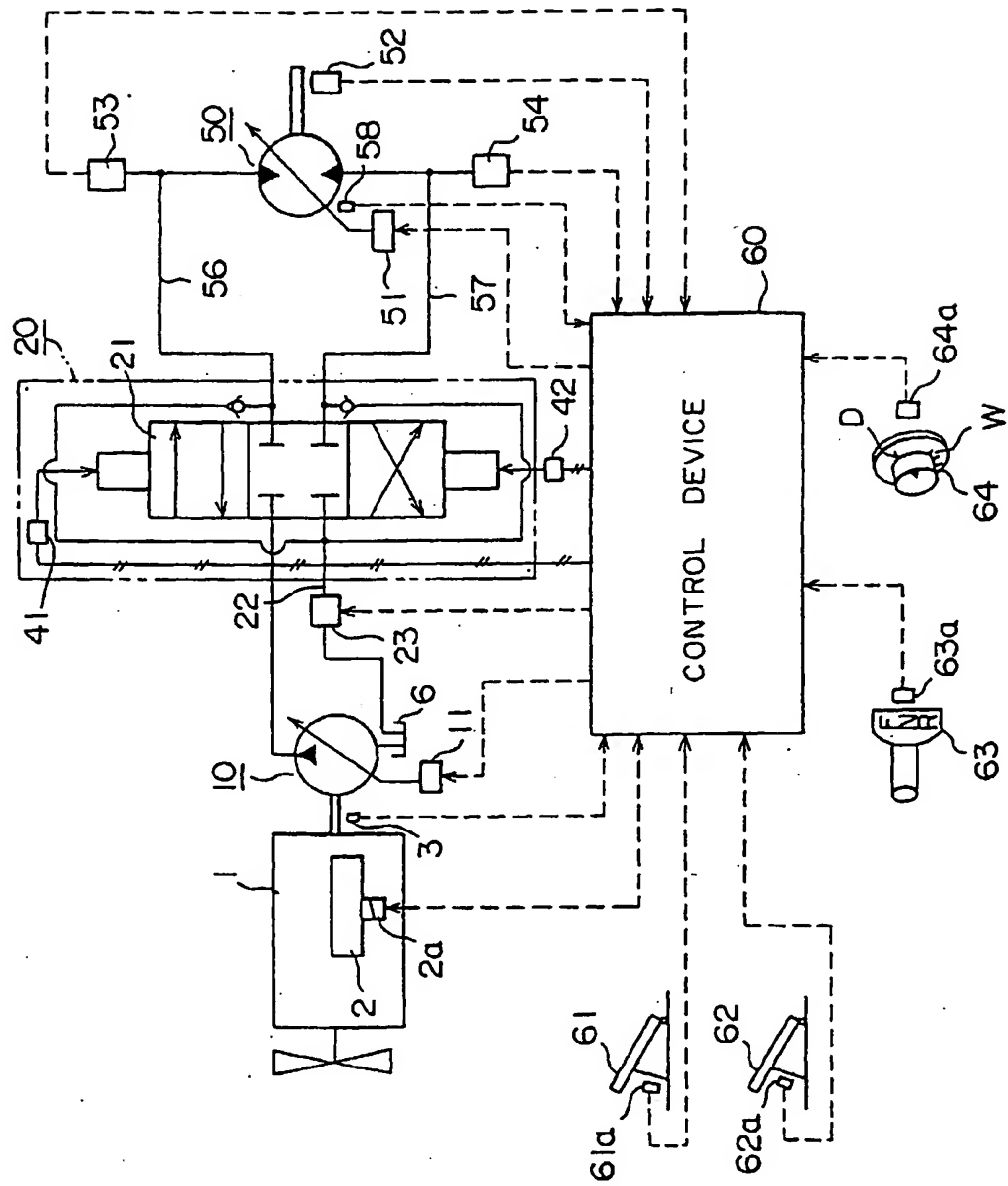


FIG. 2

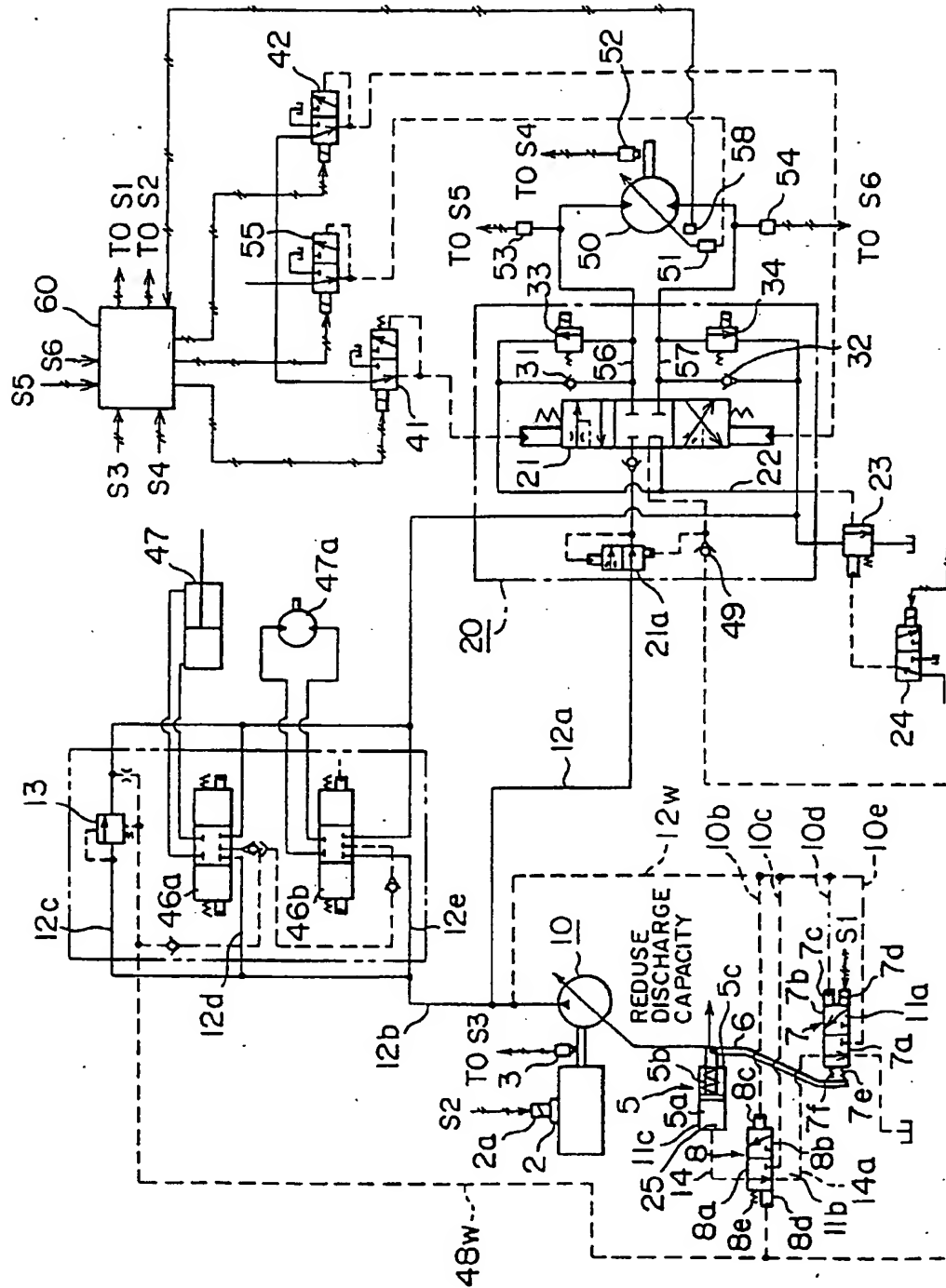


FIG. 3

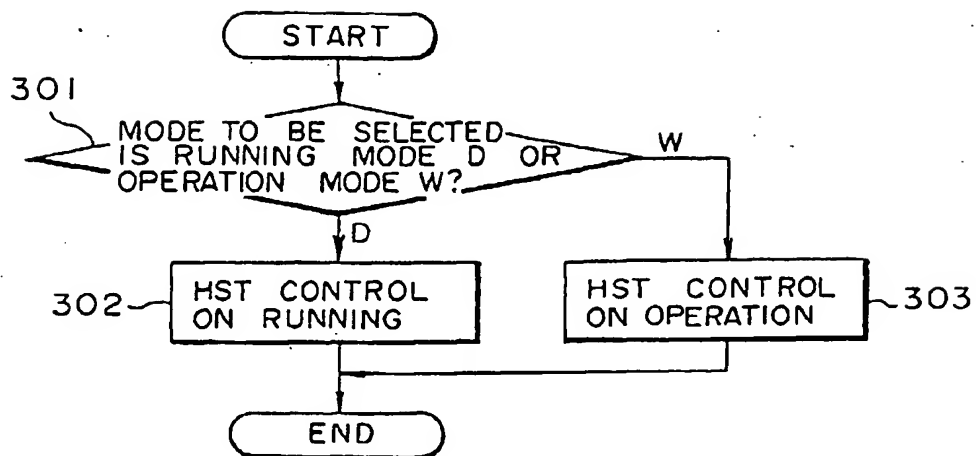


FIG. 4

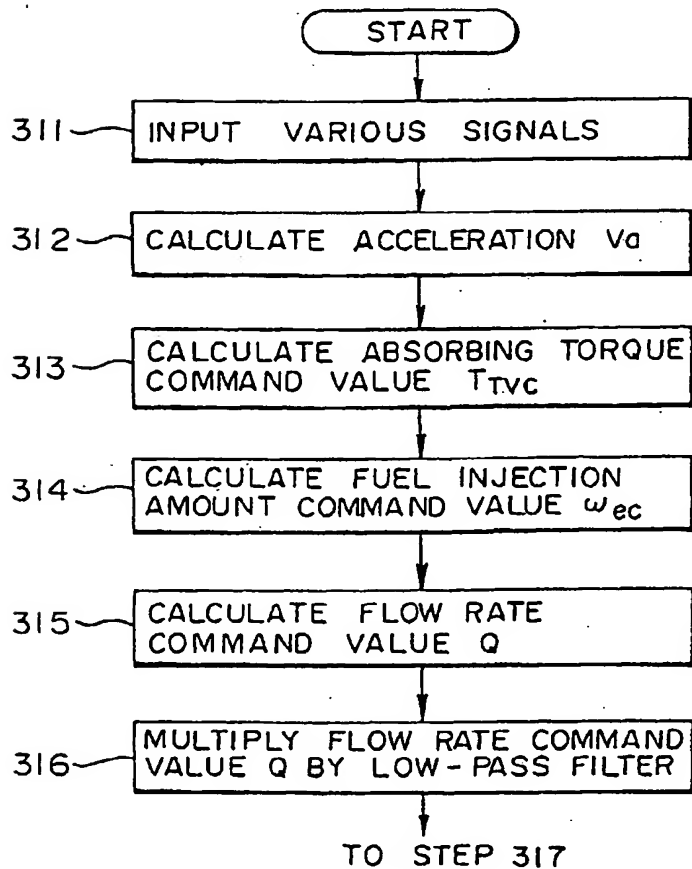


FIG. 5

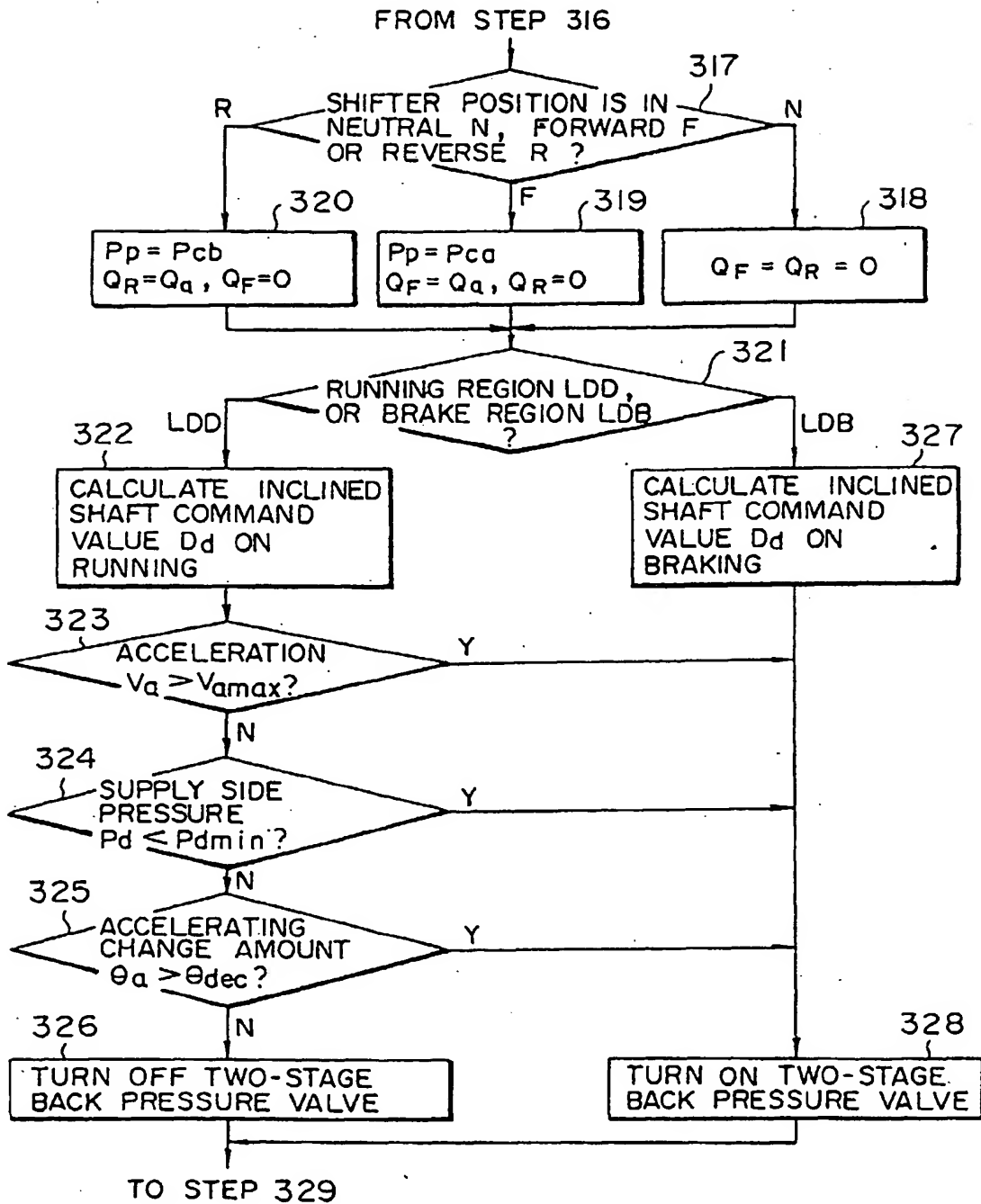


FIG. 6

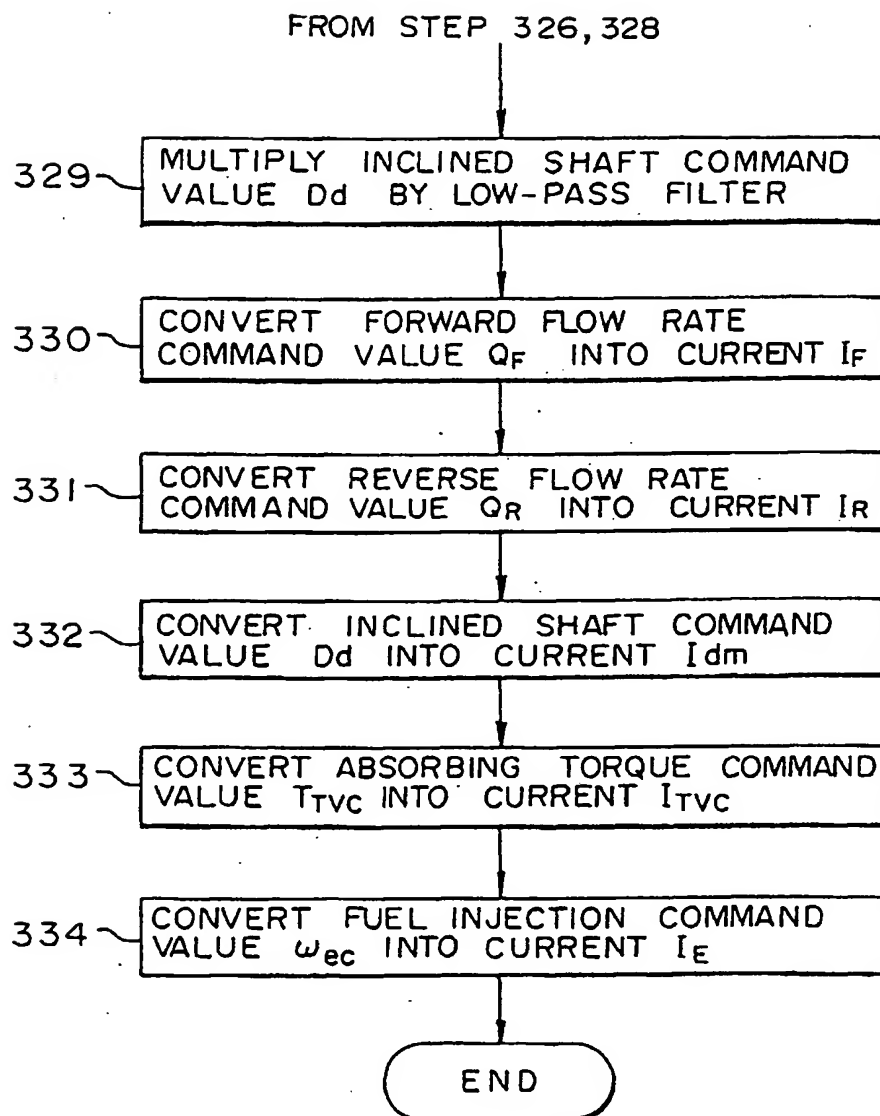






FIG. 10

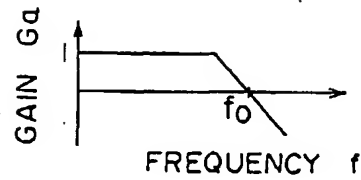


FIG. 11

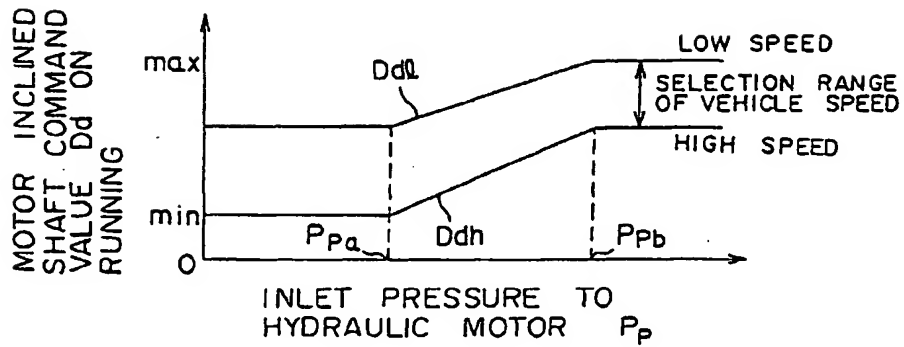


FIG. 12

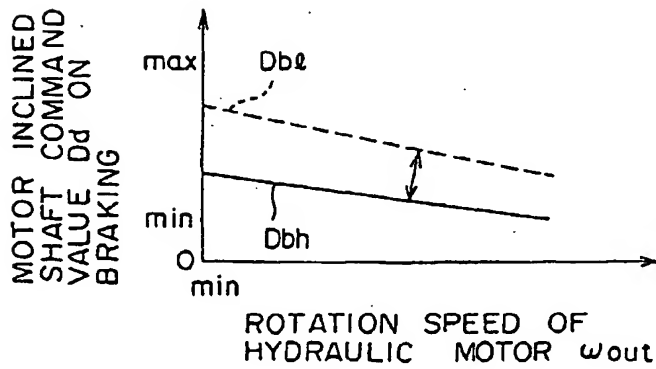


FIG. 13

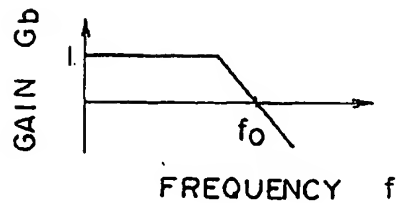


FIG. 14

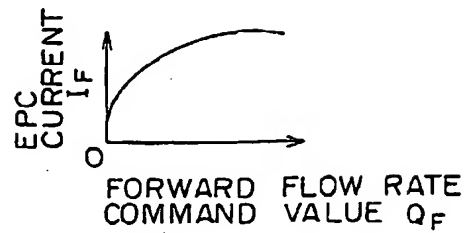


FIG. 15

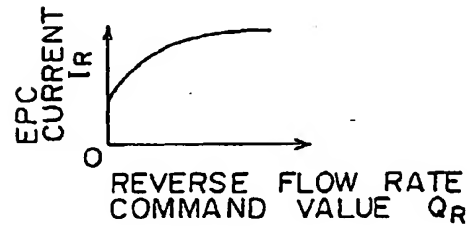


FIG. 16

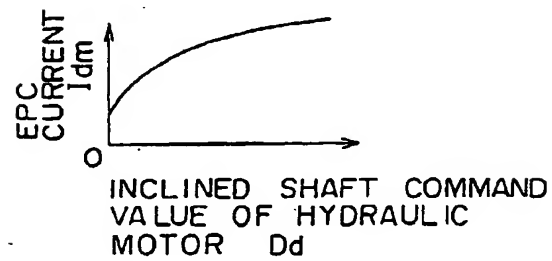


FIG. 17

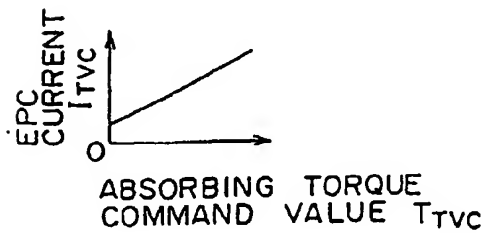


FIG. 18

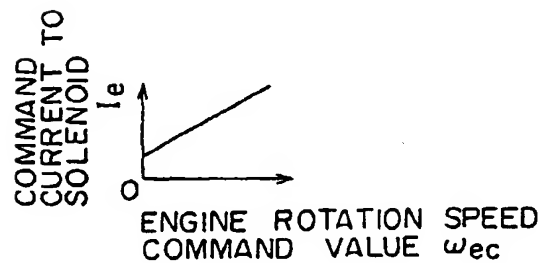


FIG. 19

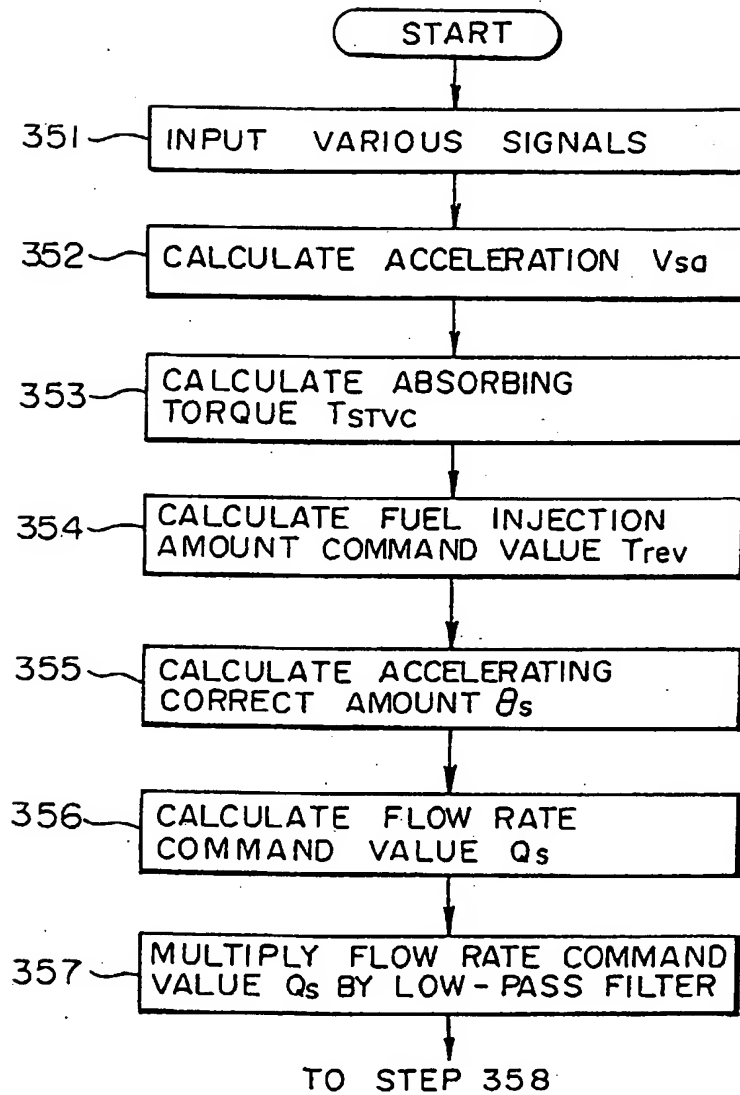


FIG. 20

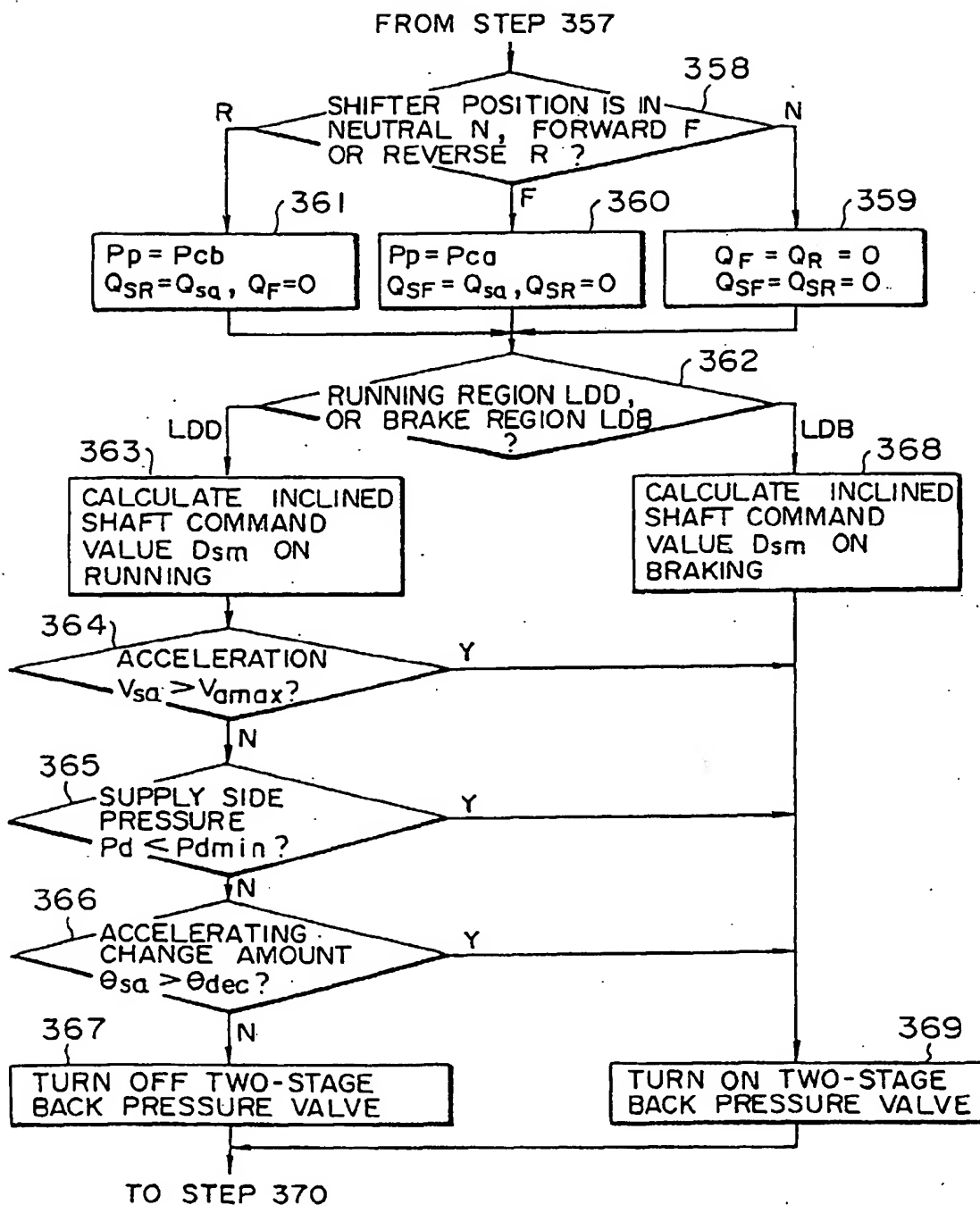


FIG. 21

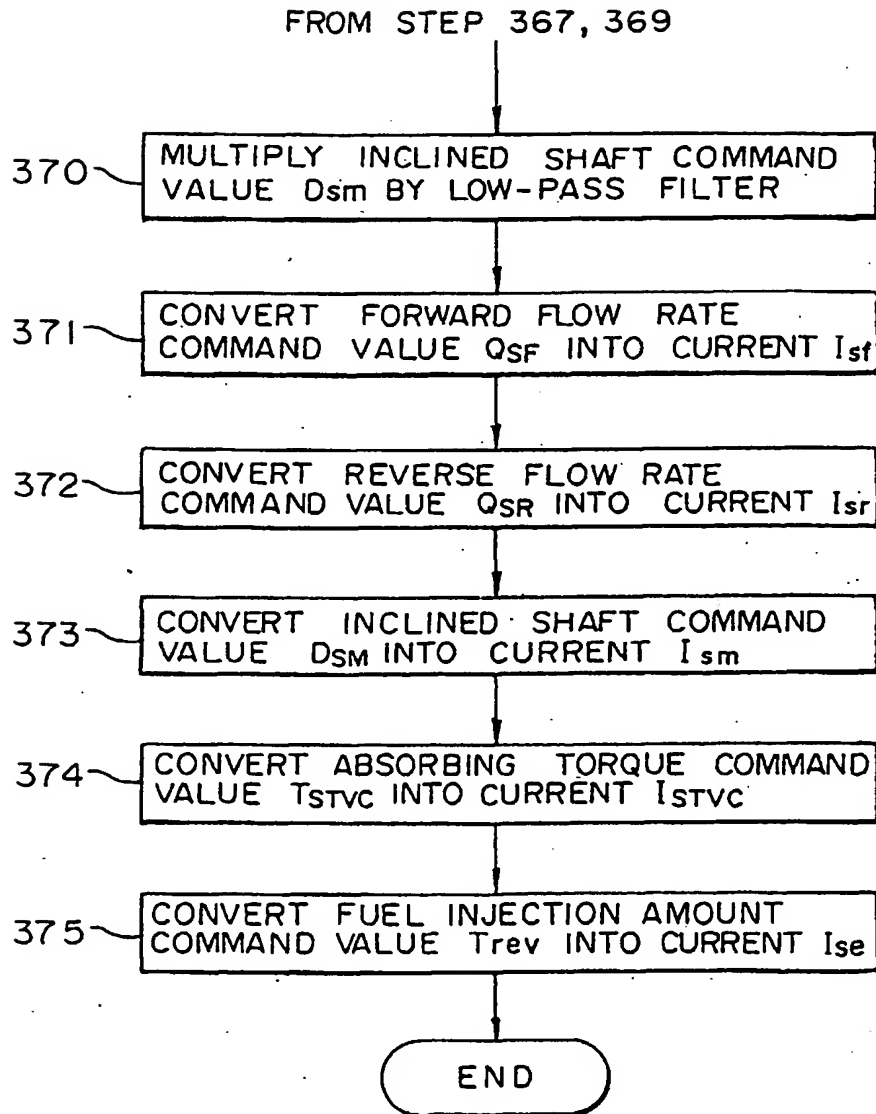


FIG. 22

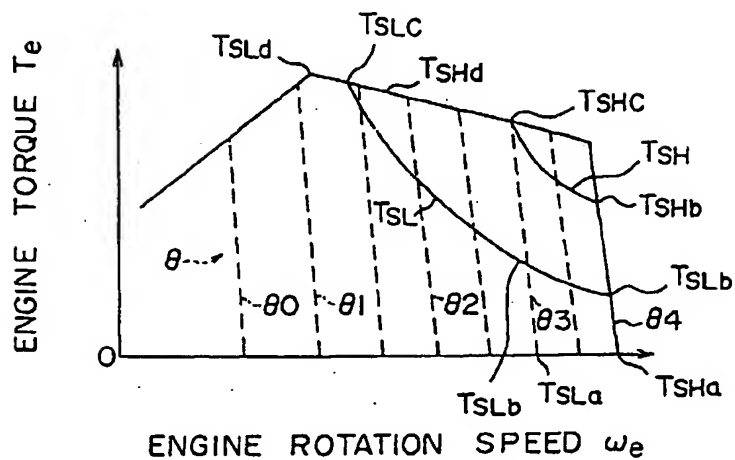


FIG. 23

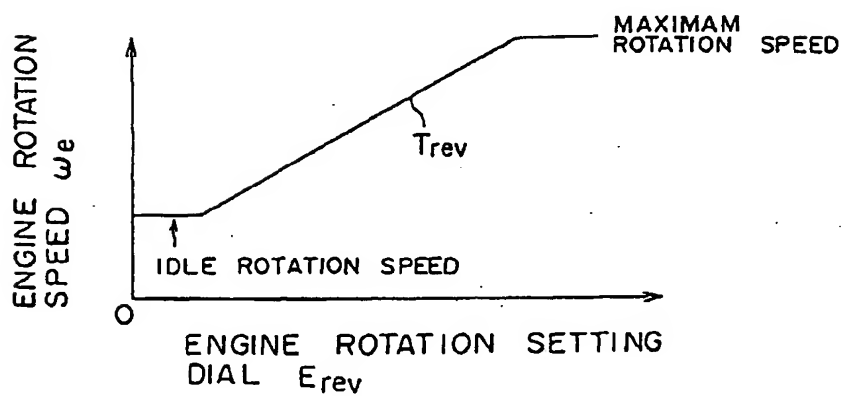


FIG. 24

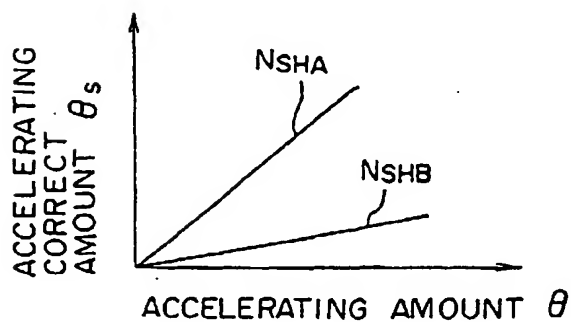


FIG. 25

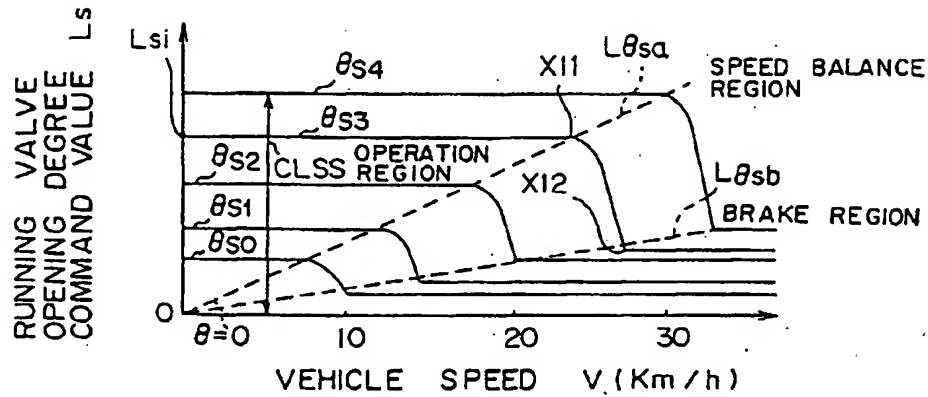


FIG. 26

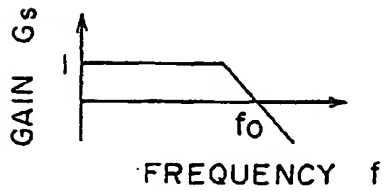


FIG. 27

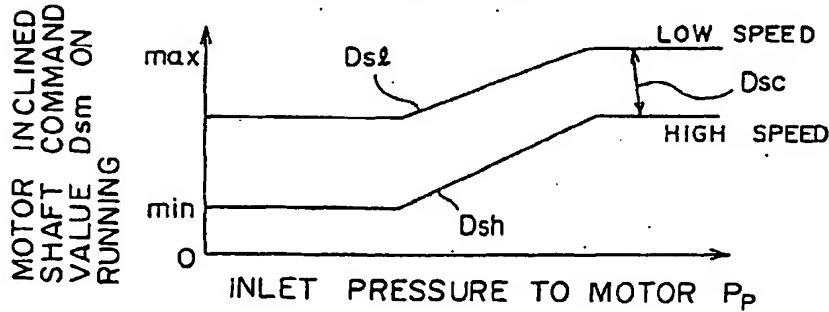


FIG. 28

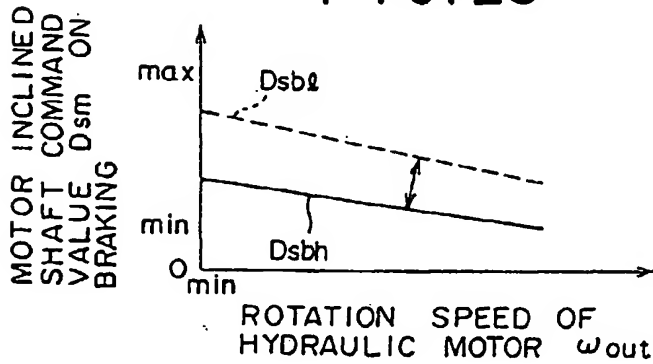




FIG. 29

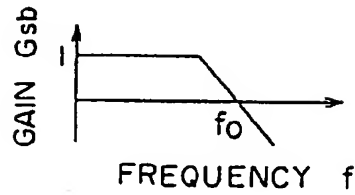


FIG. 30

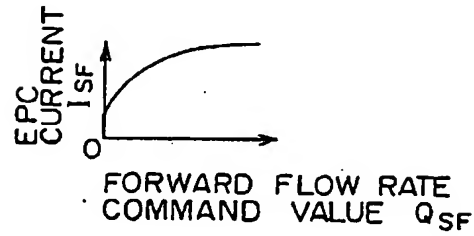


FIG. 31

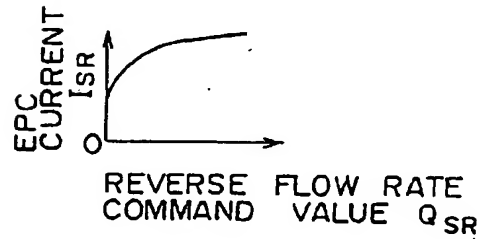


FIG. 32

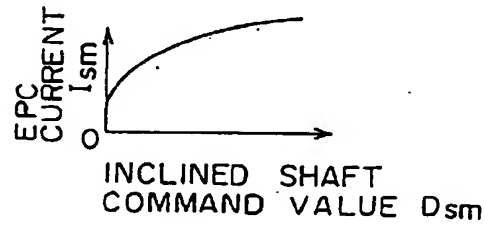


FIG. 33

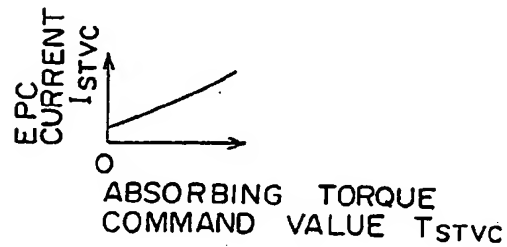


FIG. 34

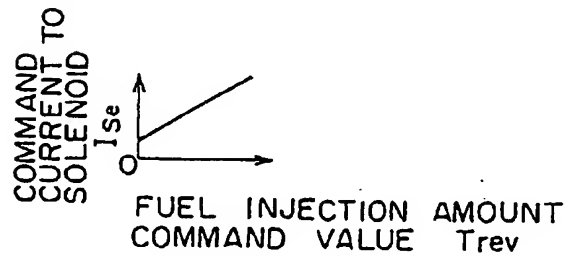


FIG. 35

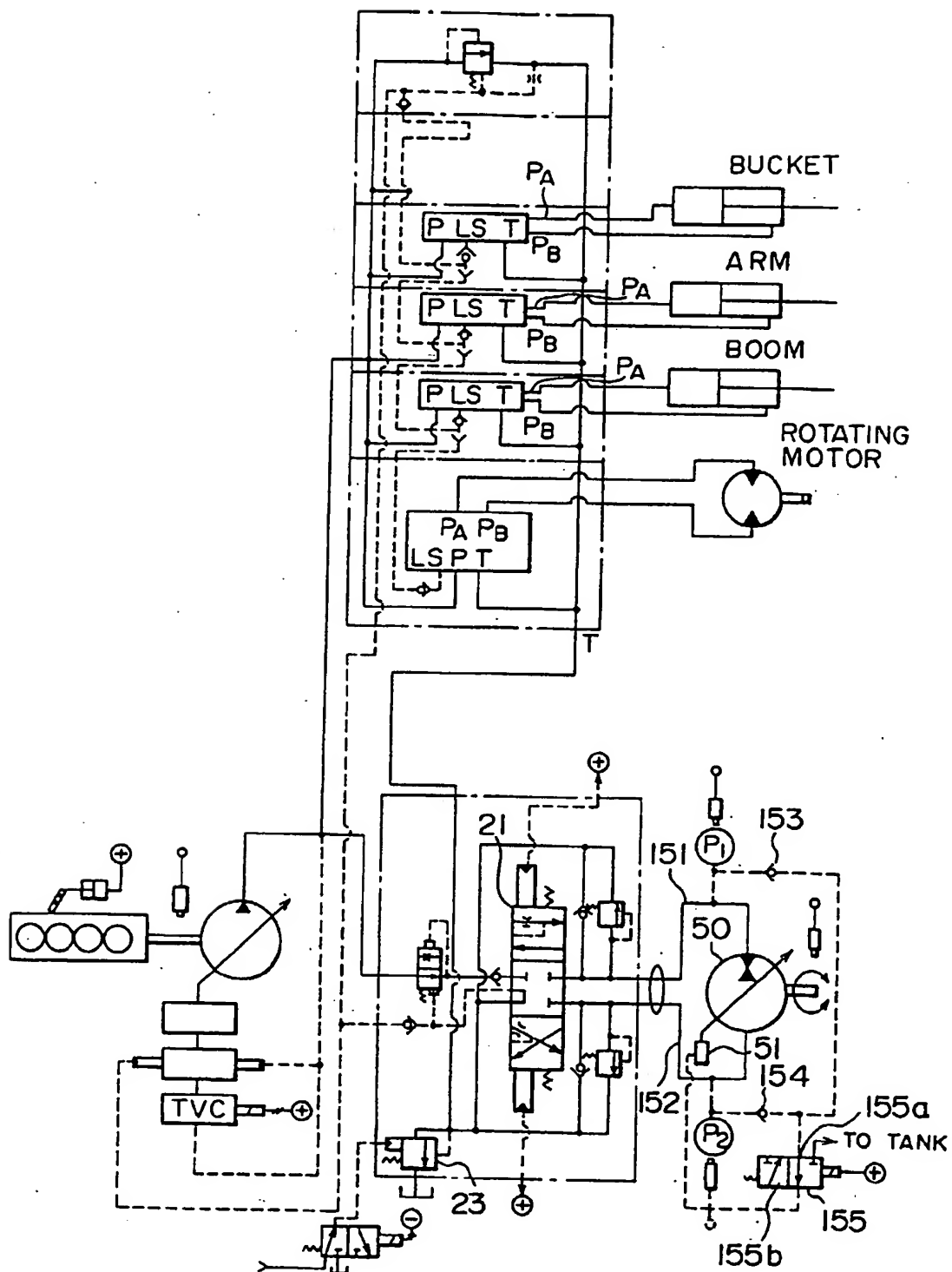


FIG. 36

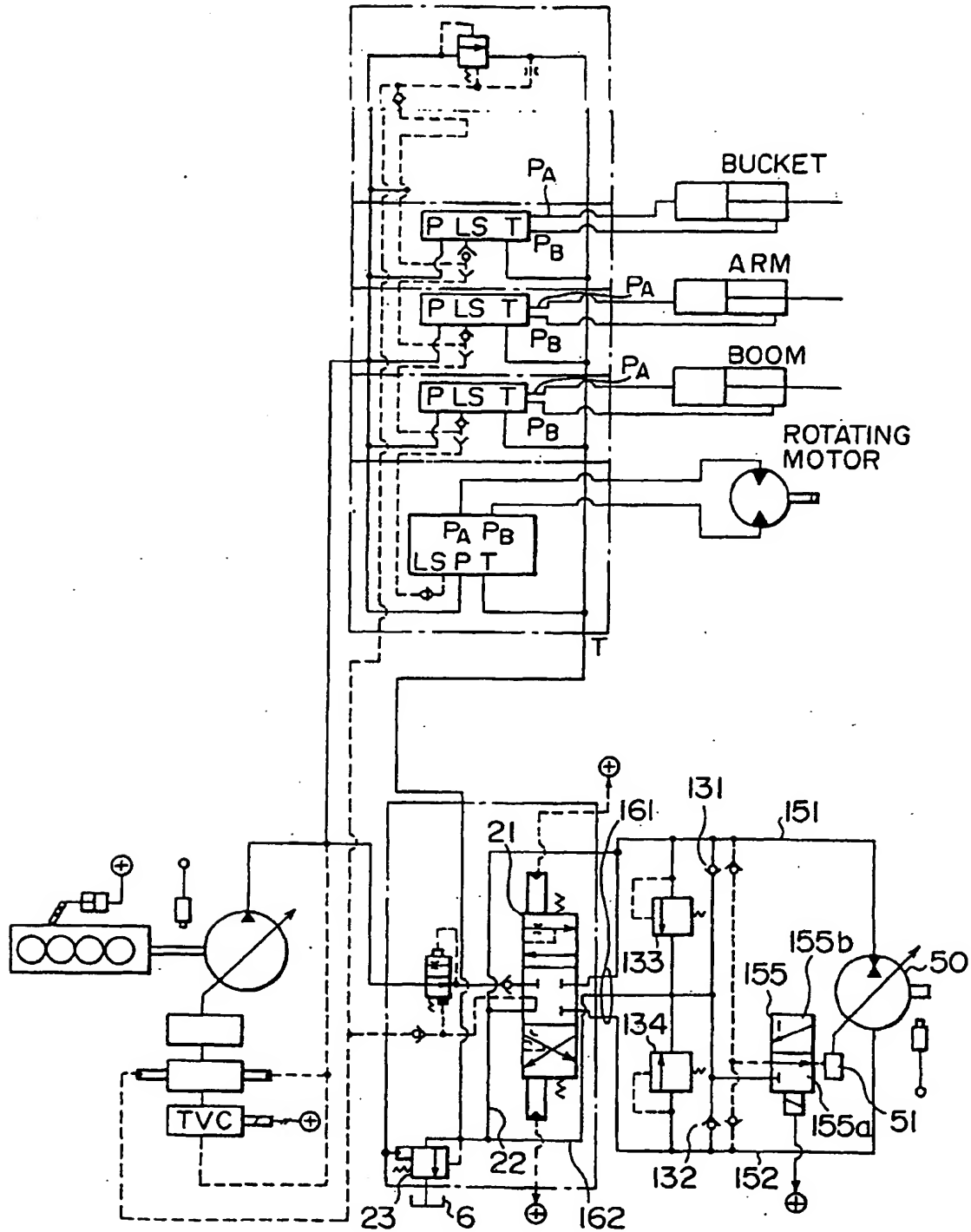


FIG. 37  
PRIOR ART

